The Design of Marine Engines and Auxiliaries

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PREFACE

THE production of a book upon marine engine design must necessarily involve the use of material from many sources. It is so difficult to determine the ultimate source of all this material that the author has not attempted the task. It is far easier to point out those portions of the book which have some degree of originality and then to make a general acknowledgment of indebtedness for the remainder.

So far as the author knows the following methods are original: the method of design (§§ 13 to 22), the method of obtaining mean bearing loads (§§ 86 to 90), the use of the mean lead in the solution of valve diagrams (§ 105), the method of designing condensers (§ 156), the method of designing turning engines (§§ 173 to 177).

In the section on Engine Balancing, although no portion of the material is original, much time and effort has been expended in correlating the work of various investigators.

The question of pressures upon main bearings will be found more extensively treated in a paper by the author in Vol. 18, Part I, of The Journal of the American Society of Naval Engineers.

The author wishes to acknowledge the kindness of the Newport News Shipbuilding and Dry Dock Company in permitting him to use certain drawings for Plates 1, 2, and 3.

E. M. BRAGG.

ANN ARBOR, MICHIGAN.
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The Design of Marine Engines and Auxiliaries

SECTION I

DETERMINATION OF CYLINDER DIMENSIONS

1. Conversion of Heat into Work. — The object of any heat engine is to convert heat into work and the working fluid is taken into the engine at a certain temperature and rejected at a lower temperature. The conversion of heat into work in the case of a steam engine is accompanied by a reduction of temperature and pressure of the working fluid, and by an increase in its volume. This reduction of temperature may be accomplished in one cylinder or in many cylinders. If the reduction takes place in one cylinder the engine is called a simple engine, or single-stage expansion engine; if the temperature is reduced in two stages the engine is called a Compound, or two-stage expansion engine; if in three stages a Triple, or three-stage expansion engine; if in four stages a Quadruple, or four-stage expansion engine.

2. Measurement of Power. — The power developed by a heat engine can be measured by the change in temperature of the working fluid and also by the change in pressure and volume. In the case of reciprocating engines it is more convenient to measure the power in the latter way and the steam indicator is used for this purpose. In designing engines it is usual to assume some theoretical relation between the pressure and volume of the working fluid and then allow for the variation of the actual from the assumed relation by means of a design factor.

3. Mean Referred Pressure. — The actual performance of the engine is determined by means of indicator cards, and for design purposes it is usual to reduce the m.e.p.'s of cards ob-

tained from different cylinders to an equivalent m.e.p. when spread over the area of the low-pressure cylinder. The sum of the equivalent m.e.p.'s for all the cylinders of an engine is called the mean referred pressure.

$$\text{m.e.p.}_H \frac{\text{HP area}}{\text{LP area}} = \text{m.r.p.}_H,$$

$$\text{m.e.p.}_M \frac{\text{MP area}}{\text{LP area}} = \text{m.r.p.}_M,$$

$$\text{m.e.p.}_L = \underline{\text{m.r.p.}_L}$$

$$\text{M.R.P.}$$

The ratio of this M.R.P. to the theoretical M.E.P. obtained from the assumed relation of pressure and volume is called the design factor.

$$F = \frac{\text{M.R.P.}}{\text{M.E.P.}}.$$

4. Design Factors. — It is usual to calculate the theoretical mean effective pressure by means of a formula involving the initial pressure (absolute) of the steam, the number of expansions, the pressure-volume ratio, and the absolute back pressure. The theoretical m.e.p. will vary, depending upon what values are used for these various quantities. In some systems of data keeping the initial pressure P_i is taken as the boiler pressure, in others as boiler pressure minus five pounds, and in others as the maximum pressure in the high-pressure cylinder. The back pressure P_b is sometimes taken as zero, in other cases as four pounds absolute, and in other cases as the actual mean back pressure in the low-pressure cylinder. The pressure-volume ratio is usually taken for convenience as p.v. = constant, although in some cases it is taken as $\text{p.v.}^{\frac{17}{16}}$ = constant. The number of expansions R used may be the nominal number, taking no account of clearance and piston rods, or it may be the actual number R_a, in which case clearances and piston rods are included. Sometimes the pressure-volume curve is drawn tangent to the point of cut-off of the high-pressure indicator card. (See Fig. 1.)

If the pressure-volume ratio is assumed to be p.v. = constant the following result will be obtained:

$$\text{M.E.P.} = P_i \frac{1 + \log_e R}{R} - P_b. \qquad (1)$$

The design factor will be

$$F_1 = \frac{\text{M.R.P.}}{P_i \frac{1 + \log_e R}{R} - P_b}. \qquad (2)$$

It follows from (2) that

$$\text{M.R.P.} = \left[P_i \frac{1 + \log_e R}{R} - P_b\right] F. \qquad (3)$$

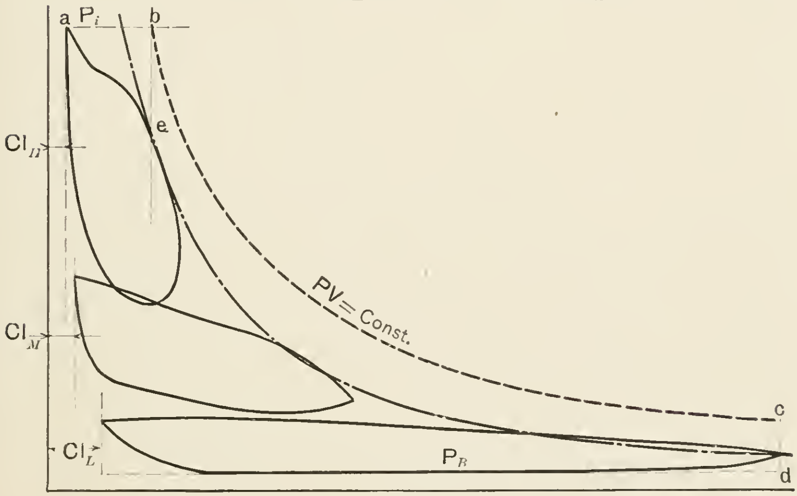


FIG. 1.

The above expression gives good results when dealing with engines having about the same P_b, but now that engines are often used in combination with a low-pressure turbine the back pressure may be 10 pounds absolute, or even higher, and more consistent results will be obtained with the following expression:

$$F = \frac{\text{M.R.P.} + P_b}{P_i \frac{1 + \log_e R_a}{R_a}}, \qquad (4)$$

$$\text{M.R.P.} = P_i \frac{1 + \log_e R_a}{R_a} F - P_b. \qquad (5)$$

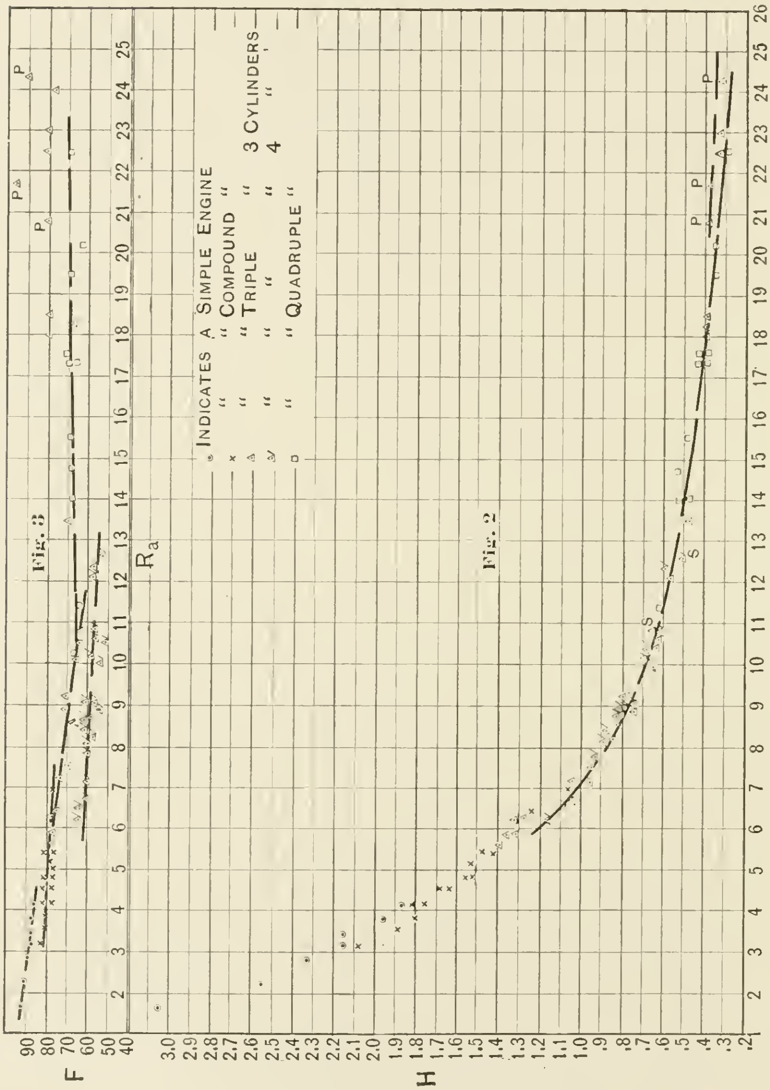


Fig. 3
Fig. 2
INDICATES A SIMPLE ENGINE
" COMPOUND "
" TRIPLE " 3 CYLINDERS
" " " 4 "
" QUADRUPLE "
R_a
F
H

FIGS. 2, 3.

The quantity M.R.P. + P_b will be designated as M.R.P_0.

In Fig. 3 will be found values of F plotted upon R_a as abscissæ. These values of F were obtained by taking for P_i the maximum absolute pressure in the high-pressure cylinder, for P_b the absolute mean back pressure in the low-pressure cylinder, and for R_a the actual number of expansions, allowing for clearances and piston rods.

A study of these factors will show that they are not by any means constant. They vary with the number of expansions and also with the initial steam pressure. There is nothing in Formula (4) which takes into account the variation of pressure for a given number of expansions. In fact, if the above equation is put in the following form,

$$F = \frac{\text{M.R.P.} + P_b}{P_i} \frac{R_a}{1 + \log_e R_a},$$

it can be seen that an equation of the form,

$$G = \frac{\text{M.R.P.} + P_b}{P_i}$$

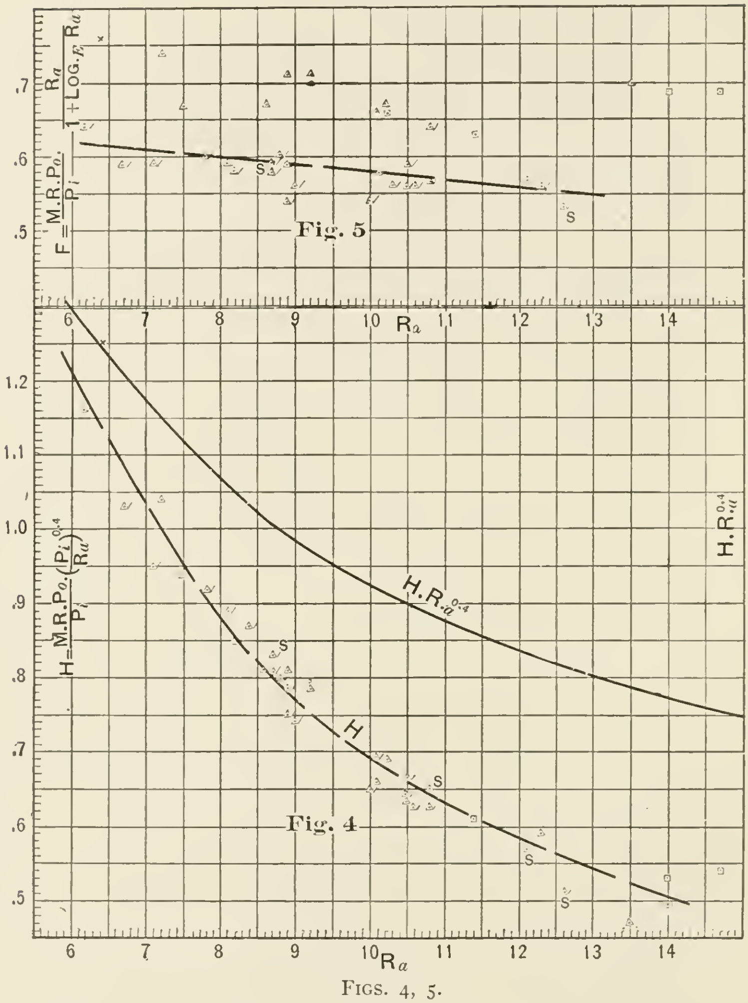
would be just as constant for any given value of R_a as is Formula (4). The term $\frac{R_a}{1 + \log_e R_a}$ merely serves to make the values of F vary less in going from one value of R_a to another.

For this reason the following expression has been used to obtain a design factor H:

$$H = \frac{\text{M.R.P.} + P_b}{P_i} \left(\frac{P_i}{R_a}\right)^{0.4}. \tag{6}$$

Values of F and H for several engines are given in Figs. 2 and 3, and portions of these figures are plotted to a larger scale in Figs. 4 and 5. It will be noticed that while the values of H vary more with R_a than do the values of F, yet for any given value of R_a the values of H are more constant.

A mean line drawn through the values of H will enable us to work back and get values of F for different initial pressures as



FIGS. 4, 5.

shown in Fig. 6. The curves in this figure give the relation between F, M.R.P.$_0$ and $\frac{P_i}{R_a}$. It will be noticed that the values of F decrease as P_i increases, and that for any given value of P_i there seems to be a minimum value of F when $R_a = 9$.

5. Conditions Affecting Design Factors. — Before proceeding to make use of F and H for design purposes it will be necessary

to know how these factors are affected by the various conditions that may exist in an engine. Our expressions for F and H contain only functions of M.R.P.$_0$, P_i, and R_a, whereas such conditions as the amount of superheat, reheating, jacketing, cut-offs used, wetness of steam, valve leakage, etc., all have their effects,

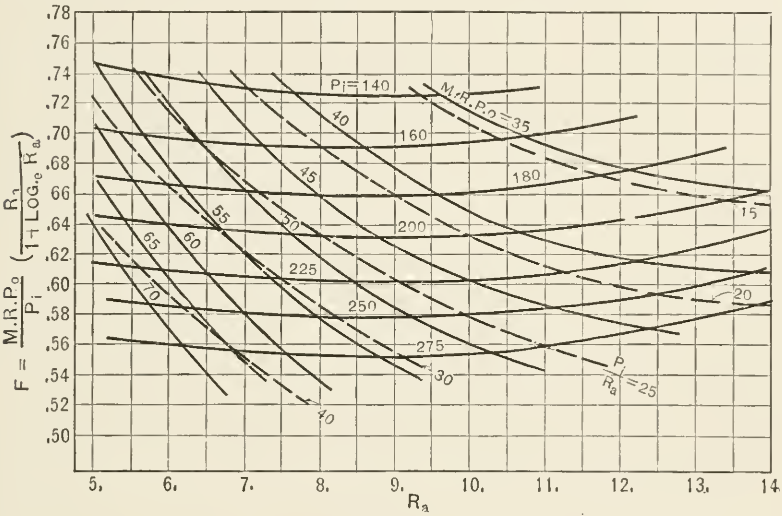


Fig. 6.

and our expressions should contain functions of all these if we wish to get *constant* values.

In the problem of engine design we must keep clearly in mind the difference between those conditions which make for maximum power and those which make for maximum economy. The two sets of conditions are not identical and most engines are a compromise between these conditions.

6. Superheated Steam. — The use of superheated steam affects the engine in three ways: it decreases pounds of steam per i.h.p.; it decreases the mean referred pressure, and so decreases the design factor; and it decreases the mechanical efficiency of the engine.

When the steam enters the cylinder it gives up a certain amount of heat to the walls. If the steam is saturated this giving up of heat will be accompanied by a precipitation of water

upon the cylinder walls, and due to the presence of this water the interchange of heat between steam and walls will be greatly accelerated. If superheated steam is used it can lose some heat without causing precipitation and the loss of heat to the cylinder walls will be greatly reduced. This condensation of steam, while detrimental to economy, is beneficial to the mechanical efficiency since the water acts as a lubricant. For this reason the steam consumption should always be referred to the brake horse-power.

Superheating the steam makes it less dense and gives leaner indicator cards; hence an engine must be larger for a given power

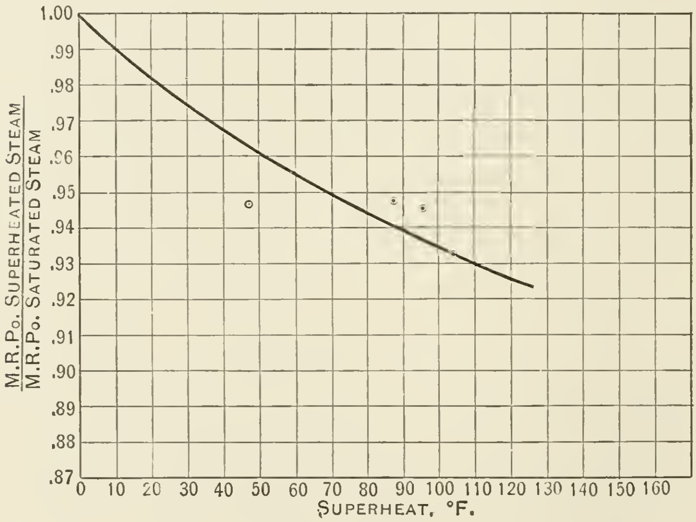


Fig. 7.

than when saturated steam is used. Fig. 7 shows the effect of superheat upon the design factor as deduced from the trials of the steam-yacht *Idalia*. In Fig. 4 it will be noticed that certain points marked *S*, indicating that superheated steam was used, are below the points for engines using saturated steam.

It is usually estimated that there is 1 per cent decrease in the pounds of steam used per i.h.p. for every 10° F. superheat, for moderate ranges of superheat. In marine practice it is generally considered best not to use more than 175° or 200° F. superheat

at the engine as troubles are experienced when the temperature of the steam exceeds 600° F.

Since with superheated steam we get leaner indicator cards and since the superheat decreases as we go towards the L.P. cylinder, the distribution of power will be different in an engine when using superheated steam from what it is when using saturated steam. The effect of superheat is to cause the L.P. cylinder to do a larger percentage of the total work.

In this connection it is interesting to note that the values of F and H for the U.S.S. *Michigan*, using steam superheated about 88°, fall below the average of those using saturated steam, as would be expected. In the case of the U.S.S. *Delaware*, however, the values are above the average, and in the case of the *Creole* are just about the same as those for saturated steam. The reason for this will be explained later.

7. Reheating. — Reheating the steam between cylinders has the effect of increasing the design factor as is shown in the case of the pumping engines in Fig. 2 which are marked P. It will also cause the steam to be drier in the M.P. and L.P. cylinders and by decreasing the lubrication will cause the mechanical efficiency to be lowered. If steam from the boiler is used as a reheating agent the total pounds of steam per b.h.p. may in some cases be increased. As in the case of superheated steam the steam consumption should be expressed in terms of the brake horse-power. Professor Weighton drew the following conclusions from experiments made upon a compound engine (Inst. of Mech. Eng., July, 1902).

"(1) The amount of steam condensed in the coils is independent of the temperature difference between heating and heated steam but depends upon the amount of heated steam used.

"(2) The influence of the reheater is beneficial in the following respects:

(*a*) Reduces the amount of condensation in receiver.
(*b*) Raises receiver pressure.
(*c*) Raises mean referred pressure.
(*d*) Increases the r.p.m.
(*e*) Increases the dryness of steam in L.P. cylinder.

"Detrimental to economy in the following:

(*a*) Lowers the mechanical efficiency of engine.

(*b*) Increases the steam per horse-power developed."

Reheating also affects the distribution of power. It increases the receiver pressures and so increases the back pressures in the H.P. and M.P. cylinders. This throws more work into the L.P. cylinder.

It is generally considered in the United States that reheating increases the economy of pumping engines.

8. Jacketing. — The effect of jacketing upon the design of the engine will depend upon the type of engine and the extent of jacketing. The following results are taken from experiments made by Professor Mellanby upon a compound engine with cylinders $11\frac{1}{2}''$ and $20''$ in diameter and a stroke of $36''$.

		H.P.	L.P.
P_i = 155 pounds (absolute),	Cut-off	= 0.25	0.52
P_b = 2.8 pounds (absolute),	Clearance	= 0.092	0.07
R.P.M. = 60,	P.S. = 360 feet per min.		

	Lbs. of steam per I.H.P.	M.R.P. in H.P. cyl., lbs.	M.R.P. in L.P. cyl., lbs.	M.R.P.$_0$	Per cent M.R.P.$_0$ in H.P.	H
No jackets	18.2	21.5	14.3	38.5	55.7	0.75
H.P. ends jacketed	17.8	21.8	13.6	38.1	57.2	0.74
H.P. ends and barrel jacketed	17.4	22	13.4	38.1	57.6	0.74
H.P. ends, barrel, and L.P. ends	17	20.9	17	40.8	49	0.77
H.P. and L.P. ends and barrels	17.3	20.2	19	42.1	47	0.81
L.P. ends and barrel	17.5	....	....	....	....	....

It will be noticed that the extent of the jacketing affected the design factor, the steam consumption, and the distribution of power. The following conclusions are drawn from experience and the results of experiments:

"All causes, superheat, late cut-offs, working noncondensing, high speed (short stroke), which tend to raise the temperature of the walls diminish the useful effect of jackets. Jackets afford a larger surface for radiation. Jackets are good for engines of intermittent action. Saturated steam is the best medium for jacketing. The rates of heat transmission when

walls are wet and when dry are as 360 to 1. Jackets are useful for slow revolution but not for quick revolution engines. Jackets are useful for compound and simple engines but are of doubtful value on Triples and Quadruples."

9. Effects of Cut-off. — The cut-off in the M.P. and L.P. cylinders will affect the total power developed, as shown by the experiments of Prof. R. L. Weighton upon a triple expansion engine:

	Cut-offs	p_i	M.R.P.
$\frac{7'' - 15\frac{1}{2}'' - 23''}{18''}$	0.67 — 0.67 — 0.67	201	28.2
	0.67 — 0.3335 — 0.67	201.5	28.4
$\frac{7'' - 10\frac{1}{2}'' - 23''}{18''}$	0.67 — 0.64 — 0.67	201	27.1
	0.67 — 0.64 — 0.335	202	28.6
	0.67 — 0.64 — 0.25	200	27.5

These experiments were carried out by Professor Weighton primarily to investigate the effect of varying cut-off upon economy and as a result of the experiments the following conclusions are given:

(See "Receiver Drop in Multiple-expansion Engines," *North-East Coast Institute of Engineers and Shipbuilders*, Vol. 16.)

"(1) For maximum economy of consumption steam must be cut off at a certain point of the stroke in the larger cylinders (I.P. and L.P.) of multiple-expansion engines. For any given cylinder this point depends solely upon the ratio between the capacities of that cylinder and the preceding cylinder (R), and is expressed as follows:

$$\frac{\text{Maximum-economy cut-off}}{\text{Stroke}} = 0.15 + \frac{1}{R}.$$

This is the principal and most important conclusion deduced from the trials.

"(2) It follows from No. 1 that the cut-off in the larger cylinders, once fixed, should never be altered, whatever may be the cut-off in the high-pressure cylinder, or the steam pressure employed. This means that automatic-expansion governors, or linking-up gear, should act upon the high-pressure cylinder only, if maximum economy at all powers is to be preserved.

"(3) The cut-off in the larger cylinders affects materially the total horse-power developed by the engines. As regards the low-pressure cylinder of triples, and the second intermediate and low-pressure cylinders of quadruples, maximum power cut-off in these cylinders coincides with the cut-off of maximum economy. As regards the second cylinder of triples and quadruples, maximum power cut-off is very considerably later than that of maximum economy. In compounds the maximum power cut-off in the low-pressure cylinder is only slightly later than that of maximum economy.

"(4) When cylinder ratio (R) is small — say from 2 to 2.5 — the cut-off in the larger cylinders may be varied considerably from that corresponding to maximum economy without any appreciable fall in economy. But when (R) is large there is no such permissible deviation without entailing as a consequence a fall in efficiency.

"(5) With given efficiency in receiver drop, smallness of cylinder ratio is conducive to smoothness of working, uniformity of turning, durability, handiness in starting and reversing, and compactness of design."

(In getting values of (R) include the effect of piston rods and clearance spaces.)

The above conclusions can be put somewhat more concisely as follows:

For economy the cut-offs should be $0.15 + \frac{1}{R}$. (Weighton.)

For maximum power the cut-off should be $\left(0.15 + \frac{1}{R}\right) 1.4$ in I.P. cylinder.

For maximum power the cut-off should be $\left(15 + \frac{1}{R}\right)$ in L.P. cylinder.

For smooth running cut-offs should be late.

For any desired distribution of power the cut-offs should be determined from the curves of Fig. 12. The service to which the engine is to be put will determine which of these conditions should be considered the most important.

In connection with the tests carried out by Professor Weighton it was shown that the mean referred pressure obtained from an engine varied considerably, dependent upon the cut-offs used in the M.P. and L.P. cylinders. In the case given above there is a variation of 6 per cent due to this cause alone. In the case of the U.S.S. *Delaware* which used superheated steam, and whose values of F and H would naturally fall below those for saturated steam, the cylinder proportions and cut-offs appear to have been so favorable that the increase due to these latter conditions was more than the decrease due to the use of superheated steam.

10. Vacuum. — The effect of vacuum upon economy was investigated in certain experiments carried out by Professor Weighton. He found that the number of pounds of steam per brake horse-power was least at a vacuum of 26 to 28 inches, but that the number of heat-units, working from H.P. steam chest to hotwell, per brake horse-power was least at 20 inches vacuum. While the horse-power of the engine is increased by increasing the vacuum, it causes the temperature of the hotwell to be lower and it is possible to carry the cooling to such a point that the coal per brake horse-power is increased.

Another point brought out by these experiments was that in going from 26 inches vacuum to 28 inches vacuum the M.R.P. was not increased by the equivalent of 2 inches of mercury but by something less than 2 inches.

It was also shown that the effect of increased vacuum was not confined entirely to the L.P. cylinder but reached back into the M.P. and H.P. cylinders with the result that the steam consumption instead of remaining constant per revolution increased as the vacuum increased. Both of these results are probably due to the increased range of temperature in the cylinders, causing an increase in the initial condensation. This effect would probably be greater with less stages of expansion and less with a greater number of stages.

11. Crank Arrangement. — Experiments were carried out by Professor Mellanby to ascertain the effect of crank arrangement in a Quadruple upon steam consumption. It was found that

with the first two and last two cranks at 180° the engine was more economical by about 3 per cent than when the cranks were at 90°. This is probably due to the fact that the 180° arrangement gives a card as shown by Fig. 9, while the 90° arrangement gives a card like Fig. 8. In the latter the lowest temperature occurs at the end of the stroke when the piston is barely moving and the full area of the walls is subjected to the low temperature

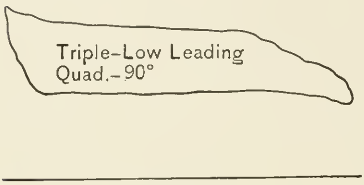


FIG. 8.

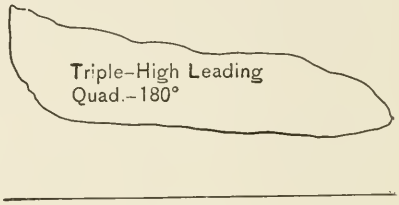


FIG. 9.

for an appreciable time. In the case of the card shown in Fig. 9, the lowest temperature occurs near midstroke when the piston is moving fastest and only about half of the area of the walls is subjected to the low temperature.

In triple-expansion engines the "high-leading" arrangement of cranks gives an indicator card similar to that of the Quadruple with cranks at 180°, while the "low-leading" arrangement gives a card similar to that of the Quadruple with the 90° arrangement.

12. Mean Effective Pressure. — The mean effective pressure employed for the production of power under any given set of conditions affects the economy of the engine very materially.

In a paper read before the Institute of Engineers and Shipbuilders, in Scotland, and printed in Vol. 50, 1906–7, Mr. R. Royds made certain statements concerning the best mean effective pressure for any given condition, and these statements have been concurred in by most of the men who have had experience in testing engines. These statements are as follows:

"(1) The higher the mean effective pressure, the lower will be the first cost of a steam engine for any given power.

"(2) For multiple-expansion unjacketed condensing engines, using saturated steam at about 165 pounds per square inch absolute in the engine cylinder, the best mean effective pressure

for normal load is from 40 to 45 pounds per square inch, referred to the L.P. cylinder, and the economy varies but slightly for a considerable range in the mean effective pressure.

"(3) For jacketed multiple-expansion condensing engines, with steam pressure as above, the best mean effective pressure is slightly lower than for unjacketed multiple-expansion condensing engines.

"(4) Non-condensing engines have a best mean effective pressure rather higher, and the variation in economy for any given range of mean effective pressure is less than for condensing engines.

"(5) For steam pressures higher than 165 pounds per square inch absolute, the best mean effective pressure is higher than from 40 to 45 pounds per square inch, and is probably as high as from 45 to 50 pounds per square inch referred to the L.P. cylinder, for triple or quadruple-expansion engines using saturated steam over 200 pounds per square inch boiler pressure.

"(6) Multiple-expansion engines using saturated steam below 165 pounds per square inch absolute have their best mean effective pressure below from 40 to 45 pounds per square inch, and this best mean effective pressure falls more rapidly with fall of steam pressure for the condensing than for the non-condensing engine.

"(7) The more economical an engine can be made, the lower is likely to be the best mean effective pressure, though not to any large extent. Hence, large engines may have a rather lower best mean effective pressure than small engines using steam at the same pressure.

"(8) Engines using highly superheated steam, so that the steam is superheated during expansion, have a best mean effective pressure lower than for engines using saturated steam, with a consequent increase in first cost for any given power. Such engines have a high thermal efficiency, and will maintain the same efficiency over a wide range of power.

"(9) The best mean effective pressure is about 35 pounds per square inch for single-cylinder condensing non-jacketed

engines, using saturated steam at about 75 pounds per square inch absolute. For other conditions the same general laws hold good as for multiple-expansion engines."

The following table shows somewhat more concisely the desirable mean referred pressure for different conditions:

TABLE 1

	Single-cylinder, 75 pounds absolute	Multiple-expansion, 165 pounds absolute	Multiple-expansion, 200 pounds absolute	Multiple-expansion, 250 pounds absolute
Unjacketed, non-condensing...	37	42–47	48–53	53–58
" condensing	35	40–45	45–50	50–55
Jacketed, condensing..........	33	38–43	42–47	47–52
Superheated slightly..........	32	36–41	40–45	45–50
" highly	30	33–38	37–42	42–47

The steam consumption tests carried out upon H.M.S. *Argonaut*, U.S.S. *Birmingham*, U.S.S. *Delaware*, and U.S.S. *Texas* agree with these conclusions. Below are given the proportions of the engines for these ships and some of the results of the tests for steam consumption.

TABLE 2

	Diam. of cylinder	Per cent of clearance	Cylinder volume ratio	Cut-off for maximum economy	Jackets	
					Liners	Ends
Argonaut H	34	25	1	...	yes	no
I......	$55\frac{1}{2}$	20	2.6	.54	yes	no
(2) *L*.....	64	15	6.65	.54	yes	no
Birmingham H ...	$28\frac{1}{2}$	26.2	1	...	no	no
I...	45	21.3	2.5	.55	no	no
(2) *L*...	62	21.1	9.6	.41	no	no
Delaware H	$38\frac{1}{2}$	16.2	1	...	yes	no
I......	57	13.4	2.15	.62	yes	yes
(2) *L*......	76	12.4	7.6	.43	yes	yes
Texas H.........	39	14.2	1	...	no	no
I.........	63	14	2.64	.53	yes	yes
(2) *L*.........	83	15	9.27	.435	yes	yes

TABLE 3

	I.H.P.	R.P.M.	Cut-offs H-I-L	Actual expansions R_a	P_i (abs.)	$\frac{P_i}{R_a}$	M.R.P.	Vacuum, inches	Superheat at engine	Steam consumption, lbs.
Max. Power										
Argonaut......	9,390	127.5	71–67–46	8.7	245	28.3	47.3	26.0	0°	15.75
Birmingham...	7,385	192.2	79–75–60	11.35	233	20.5	34.8	27.5	0°	17.4
Delaware......	14,716	128.2	86–80–62	8.7	263	30.2	52.4	26.3	54.6°	13.4
Texas..........	14,213	124.4	77–78–62	11.6	261.1	22.5	43.8	26.7	37 7°	14.0*
Medium Power										
Argonaut......	6,906	116.3	73–67–46	8.5	189	22.3	38	26.5	0°	16.2
Birmingham...	5,431	173.3	79–75–60	11.35	194	17.1	28.5	27.7	0°	16.8
Delaware......	8,784	108.8	86–70–38	8.7	186	21.4	36.9	26.9	50.5°	12.7
Texas..........	9,489	110.3	61–67–62	14.1	249	17.7	33.0	26.7	38.1	13.2'
Low Power										
Argonaut......	1,933	76.3	73–67–46	8.5	77	9.1	16.2	26.4	0°	17.7
Birmingham...	1,565	113.2	79–75–60	11.35	95.5	8.4	12.6	26.0	0°	16.9
Delaware.......	2,049	68	86–70–38	8.7	78.5	9	13.8	27.7	36.3°	15.1
Texas..........	2,339	69.2	54–63–54	15.3	106.6	7	12.9	26.3	36.3	13.6*

* The proportion of the total steam consumed by the main engine was assumed to be the same in the case of the *Texas* as in the case of the *Delaware.*

TABLE 4

H.M.S. *Argonaut, Starboard Engine*

R.P.M.	Pressure at engine gage	Actual expansion R_a	I.H.P.	M.R.P.	Vacuum, inches	Pounds of steam per I.H.P.
74.7	168	16.1	1907	16.4	25.5	16.26
75	142	13.5	1922	16.4	25.5	16.68
76.7	78	9.3	1929	16.1	26.1	17.58
76	68	8.5	1931	16.3	26.2	17.72

In Fig. 10 the steam consumption has been plotted upon mean referred pressure as abscissæ, and it is quite evident that the *Birmingham's* engines, developing their maximum power with a mean referred pressure of 35 pounds per square inch, are not as economical as the engines of the *Delaware* which used a mean referred pressure of over 52 pounds per square inch. The engines of the *Texas*, using a mean referred pressure of about 44 pounds per square inch, are not as economical as the *Delaware's* at full power, although more economical at low power. The greater economy of the *Texas* at low power is probably due to the use of higher steam pressure. Table 4 taken from the trials

of the *Argonaut* shows the effect of increasing the steam pressure at reduced powers.

The *Birmingham's* engines would have been much more economical at maximum power if superheated steam had been used and if the cylinders had been of such a size that the mean referred pressure had been 50 pounds per square inch instead of 36 pounds. This would have called for cylinders of about

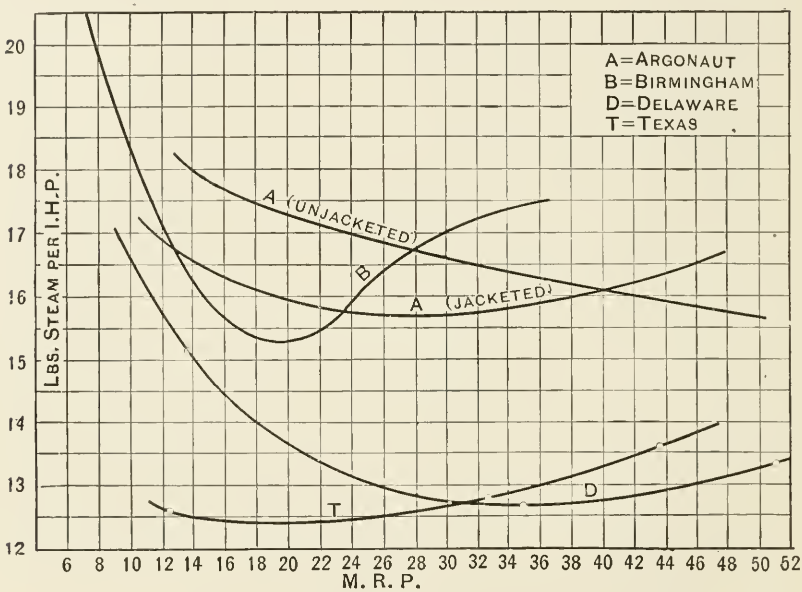


FIG. 10.

31 inches–48 inches (2) 54 inches instead of $28\frac{1}{4}$ inches–45 inches –(2) 62 inches as used.

In the case of naval engines the economy at reduced power is a matter of great importance, as the cruising speed requires only a small portion of the maximum power. Under these conditions, if it is desired to keep the cut-off in the H.P. cylinder unaltered, an engine which develops its maximum power with mean referred pressures about 20 per cent less than those given in Table 1 will be more economical at reduced power, but the economy at maximum power will suffer. If the cut-off in the H.P. cylinder is shortened at low powers to enable the power to be developed

with as high a steam pressure as possible, the engine can still use the higher mean referred pressures for maximum power and be economical at both maximum and low powers.

In the case of engines for merchant ships the economy at full power is of most importance and the larger mean referred pressures should be used.

A consideration of the effect of these varying conditions will show that it is hopeless to expect such a simple expression as (2) or (6) to give constant values. The results are more constant with (6), however, and that formula will be used hereafter to give the relation between P_i, M.R.P.$_0$, and R_a.

13. Size of L.P. Cylinder. — Since the mean referred pressure with which an engine develops its power affects the steam economy so materially it is best to start the design by assuming a value which seems desirable under the conditions. The quantities with which we will start the design will be: indicated horse-power, piston speed, boiler pressure, mean referred pressure, and back pressure in L.P. cylinder. The area of the L.P. cylinder will be

$$A_L = \frac{\text{I.H.P.} \times 33{,}000}{\text{P.S.} \times \text{M.R.P.}}, \tag{7}$$

I.H.P. = indicated horse-power,
P.S. = piston speed in feet per minute suitable for type of engine,
M.R.P. = mean referred pressure suitable for type of engine (see Table 1).

14. Number of Expansions. — The curves of Figs. 2, 4, and 6 enable us to determine the number of expansions which will give this mean referred pressure. These curves are plotted for saturated steam upon values of M.R.P.$_0$, or M.R.P. + P_B.

Since $$H = \frac{\text{M.R.P.}_0}{P_i}\left(\frac{P_i}{R_a}\right)^{0.4}, \tag{6}$$

$$H = \frac{\text{M.R.P.}_0}{P_i^{0.6}} \times \frac{1}{R_a^{0.4}},$$

or $$H \times R_a^{0.4} = \frac{\text{M.R.P.}_0}{P_i^{0.6}} \text{ (saturated steam).} \tag{8}$$

The assumed values of M.R.P. and P_B enable us to get the value of M.R.P.$_{\cdot 0}$. The boiler pressure is known and from this P_i, which is the initial pressure absolute in the H.P. cylinder, can be found. The drop in pressure from the boiler to the H.P. cylinder will depend upon the length of piping and speed of steam. At maximum power this drop usually varies from 15 to 30 pounds.

If the curve in Fig. 4 marked H is accepted as giving average values of the factor H, the value of R_a can be found directly from the curve marked H.$R_a^{0.4}$.

15. Size of H.P. Cylinder. — The size of the H.P. cylinder can be found when the value of R_a is determined.

$$R_a = \frac{\text{final volume of steam}}{\text{initial volume of steam}}$$

$$= \frac{(A_L - x)(1 + Cl_L)}{(A_H - x)(C_H + Cl_H)}.$$

A_L = area of L.P. cylinder.
A_H = area of H.P. cylinder.
x = $\frac{1}{2}$ area of piston rod.
Cl_L = clearance volume of L.P. cylinder expressed as a fraction of the stroke.
C_H = cut-off in H.P. cylinder expressed as a fraction of the stroke.
Cl_H = clearance volume of H.P. cylinder.

$$A_H = \frac{(A_L - x)(1 + Cl_L)}{R_a (C_H + Cl_H)} + x. \tag{9}$$

16. Clearances. — The values of Cl_H, Cl_L, and C_H will have to be assumed. It is good practice to keep the clearances as small as possible, and by making the ports horizontal, using fairly high piston speeds, and keeping the valves close to the cylinder, they can usually be made about as follows:

Piston valves	Flat valves
$Cl_H = 0.16$	....
$Cl_M = 0.13$	0.10
$Cl_L = 0.12$	0.09

17. Cut-offs. — C_H is usually from 0.7 to 0.85. The smaller C_H is made, the larger will be the size of the H.P. cylinder and the smaller will be the ratio between the H.P. and M.P. cylinder volumes for a given R_a. This will cause the economical cut-off in the M.P. cylinder to be more nearly equal to C_H. Fig. 11 shows the cylinder ratios which must obtain in order that the cut-off may be the same in all cylinders. There is no particular virtue in having all cut-offs the same, but the engine will run smoother and the valve diagram will work out better if the cut-offs are at least 0.65.

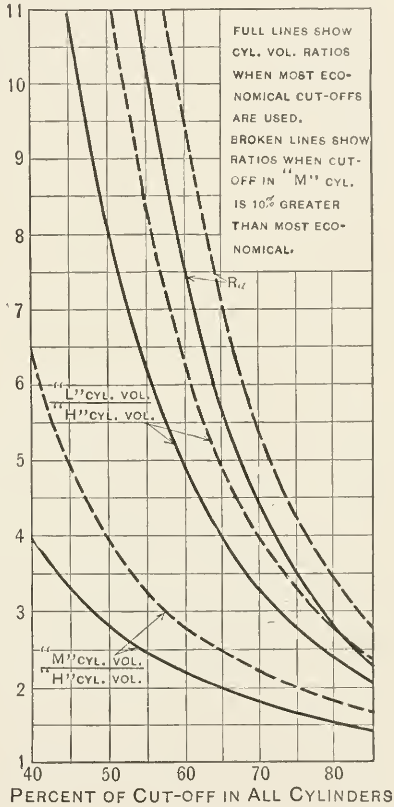


FIG. 11.

18. Sizes of Intermediate Cylinders. — The size of the L.P. cylinder having been obtained from the power to be developed, and the size of the H.P. from the number of expansions, we can place between these two cylinders as many others as we may think necessary. The sizes of these intermediate cylinders are usually made in accordance with the following:

M.P. cylinder of Triple $$\frac{A_M}{A_H} = \left(\frac{A_L}{A_H}\right)^{\frac{1}{2}}. \qquad (10)$$

1st M.P. of Quadruple $$\frac{A_{1M}}{A_H} = \left(\frac{A_L}{A_H}\right)^{\frac{1}{3}}. \qquad (10A)$$

2nd M.P. of Quadruple $$\frac{A_{2M}}{A_H} = \left(\frac{A_L}{A_H}\right)^{\frac{2}{3}}. \qquad (10B)$$

1st M.P. of Quintuple $$\frac{A_{1M}}{A_H} = \left(\frac{A_L}{A_H}\right)^{\frac{1}{4}}.$$

2nd M.P. of Quintuple $\dfrac{A_{2M}}{A_H} = \left(\dfrac{A_L}{A_H}\right)^{\frac{1}{2}}.$

3rd M.P. of Quintuple $\dfrac{A_{3M}}{A_H} = \left(\dfrac{A_L}{A_H}\right)^{\frac{3}{4}}.$

etc.

The M.P. cylinder is sometimes made smaller than the above, but in doing so the economical cut-off in the L.P. is made very short.

19. Stroke. — The stroke of the engine is usually made equal to, or a little greater than, the diameter of the M.P. cylinder. It is usually made some one of the following quantities: — 24″–27″–30″–33″–36″–39″–42″–45″–48″–51″–54″–60″–66″–72″. Naval engines rarely use a stroke greater than 48 inches.

20. Superheat Factor. — When superheated steam is to be used the same methods will be employed as for saturated steam, but another factor will have to be introduced into Formula (8).

It can be seen from Fig. 7 that the mean referred pressure obtained from steam of a given pressure decreases as the amount of superheat increases. In Fig. 4 there would be a series of curves for H under the curve for saturated steam, each curve for a particular degree of superheat. Instead of drawing these curves and taking the value of H from them we can introduce into Formula (8) a factor obtained from Fig. 7. This equation will become

$$s \cdot H \cdot R_a^{0.4} = \frac{\text{M.R.P.}_0}{P_i^{0.6}}$$

$$\therefore \quad H \cdot R_a^{0.4} = \frac{\text{M.R.P.}_0}{P_i^{0.6} s}. \qquad (11)$$

s is a factor obtained from Fig. 7.

The same M.R.P.$_0$ can be obtained from superheated steam as from saturated steam if a smaller number of expansions is used. In discussing the best M.R.P. to be used under different conditions, it was pointed out that a smaller M.R.P. could be used to advantage with superheated steam. This smaller M.R.P. would call for a larger number of expansions. The net result of decreasing the M.R.P.$_0$ in the numerator of Formula

(11) and using a factor s, less than unity, in the denominator is to call for a slightly larger number of expansions when superheated steam is used.

21. Distribution of Power. — The cut-offs used in the M.P. and L.P. cylinders will affect the steam economy of the engine, the amount of power developed, the distribution of power, and the smoothness of running. The methods for determining the cut-offs which will give maximum economy and maximum power have been given above. The distribution of power that will result from given cut-offs, or the cut-offs to give a certain distribution of power, can be found by means of Fig. 12. These curves give the mean of the results obtained from engine trials. The abscissæ are ratios of the volumes in the M.P. and L.P. cylinders at cut-off to the volume in the H.P. cylinder at cut-off. The ordinates are the portions of the total work developed in the preceding cylinder or cylinders.

It will be noticed that the curves give zero work for a volume ratio of 0.5. There are no trial results from which this part of the curve can be determined and the curves were made to pass through that point somewhat arbitrarily. Theoretically there would be zero work done in the H.P. cylinder if the volume at cut-off in the second cylinder is the same as that in the H.P. since the steam would be brought back to the original volume and pressure. There would be, however, a considerable amount of initial condensation in the second cylinder, especially if the cut-off is short, which would cause the pressure at cut-off in that cylinder to be less than it was in the H.P. cylinder, and a certain amount of work would be done. It was arbitrarily assumed that zero work would be done when the cut-off volume in the second cylinder was one-half that of the H.P. cylinder cut-off volume.

22. Example of Design. — The method of design will be illustrated by means of an example. Suppose that 30,000 I.H.P. is to be developed by two reciprocating engines and two low-pressure turbines, the turbines to take steam from the low-pressure cylinders at about 17 pounds absolute. The steam will be developed at a pressure of 275 pounds gage and with 90° of

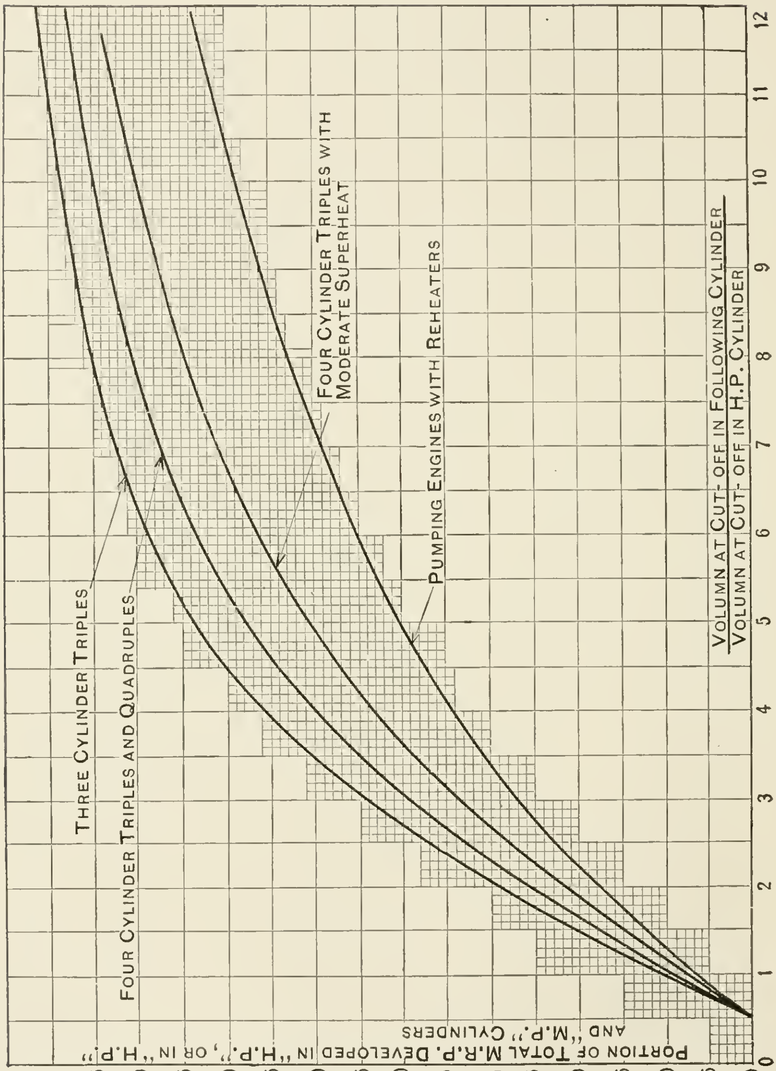

THREE CYLINDER TRIPLES
FOUR CYLINDER TRIPLES AND QUADRUPLES
FOUR CYLINDER TRIPLES WITH MODERATE SUPERHEAT
PUMPING ENGINES WITH REHEATERS
VOLUMN AT CUT-OFF IN FOLLOWING CYLINDER / VOLUMN AT CUT-OFF IN H.P. CYLINDER
PORTION OF TOTAL M.R.P. DEVELOPED IN "H.P.", OR IN "H.P." AND "M.P." CYLINDERS
0 1 2 3 4 5 6 7 8 9 10 11 12
0 .05 .10 .15 .20 .25 .30 .35 .40 .45 .50 .55 .60 .65 .70 .75 .80

FIG. 12.

superheat. A drop of 20 pounds in pressure and 30° in superheat will be allowed between boilers and cylinder.

I.H.P. = 7500, $P_i = 270$ pounds absolute, $P_b = 18$ pounds absolute,

P.S. = 1000 feet per minute, $Cl_H = 0.12$, $Cl_M = 0.11$, $Cl_L = 0.10$, $C_H = 0.75$.

From Table 1 the M.R.P. for best economy will be about 48 pounds.

$$\text{M.R.P.}_0 = 48 + 18 = 66 \text{ pounds.}$$

From Fig. 7 the superheat factor = 0.95.

$$\therefore \quad H \cdot R_a^{0.4} = \frac{66}{270^{0.6} \times 0.95} = 2.41,$$

$$R_a = 6.35 \text{ (see Fig. 4)},$$

$$\text{net } A_L = \frac{7500 \times 33{,}000}{1000 \times 48} = 5150 \text{ square inches.}$$

If two low-pressure cylinders are used, net A_L = 2575 square inches each.

$$\frac{(\text{net } A_L)\,(1 + Cl_L)}{(\text{net } A_H)\,(C_H + Cl_H)} = 6.35 = \frac{5150 \times 1 \cdot 1}{0.87\,(\text{net } A_H)}.$$

$$\therefore \quad \text{net } A_H = 1025 \text{ square inches,}$$

$$\text{net } A_M = \sqrt{5150 \times 1025} = 2300 \text{ square inches.}$$

The piston rods will be about 6 inches diameter and the half sectional area of each rod equals about 15 square inches.

$A_H = 1025 + 15 = 1040$, $D_H = 36.3''$, use $36\frac{1}{2}''$,
$A_M = 2300 + 15 = 2315$, $D_M = 54.3''$, use $54\frac{1}{2}''$,
$A_L = 2575 + 15 = 2590$, $D_L = 57.4''$, use $57\frac{1}{2}''$.
Stroke = 48 inches.

A complete investigation of the effects of the cut-off in the M.P. and L.P. cylinders upon the distribution of power can be made by calculating the distribution for 30 per cent, 50 per cent, 70 per cent, and 90 per cent cut-offs, and plotting the results.

30 *per cent cut-off in M.P. cylinder*

$$\frac{(0.30 + 0.11)\,2319}{(0.75 + 0.12)\,1032} = 1.06.$$

According to the curve for moderate superheat in Fig. 12 this cut-off volume ratio will cause 8.5 per cent of the total work to be developed in the H.P. cylinder.

50 *per cent cut-off in M.P. cylinder*

$$\text{Cut-off volume ratio} = 1.06\frac{0.61}{0.41} = 1.58.$$

Per cent of total work = 16.

70 *per cent cut-off in M.P. cylinder*

$$\text{Cut-off volume ratio} = 1.06\frac{0.81}{0.41} = 2.09.$$

Per cent of total work = 23.

90 *per cent cut-off in M.P. cylinder*

$$\text{Cut-off volume ratio} = 1.06\frac{1.01}{0.41} = 2.6.$$

Per cent of total work = 29.3.

30 *per cent cut-off in L.P. cylinder*

$$\text{Cut-off volume ratio} = \frac{(0.30 + 0.10)\,5165}{0.87 \times 1032} = 2.30.$$

Per cent of total work = 25.6.

50 *per cent cut-off in L.P. cylinder*

$$\text{Cut-off volume ratio} = 2.30\frac{0.60}{0.40} = 3.46.$$

Per cent of total work = 38.3.

70 *per cent cut-off in L.P. cylinder*

$$\text{Cut-off volume ratio} = 2.30\frac{80}{40} = 4.60.$$

Per cent of total work = 47.7.

90 *per cent cut-off in L.P. cylinder*

$$\text{Cut-off volume ratio} = 2.30\frac{100}{40} = 5.75.$$

Per cent of total work = 55.

Curves *A* in Fig. 13 are plotted from these results. The cut-offs for best economy will be as follows:

$$\text{M.P. cylinder, } R = \frac{2319 \times 1.11}{1032 \times 1.12} = 2.23,$$

$$0.15 + \frac{1}{2.23} = 0.60:$$

$$\text{L.P. cylinder, } R = \frac{5165 \times 1.1}{2319 \times 1.11} = 221,$$

$$0.15 + \frac{1}{2.21} = 0.605.$$

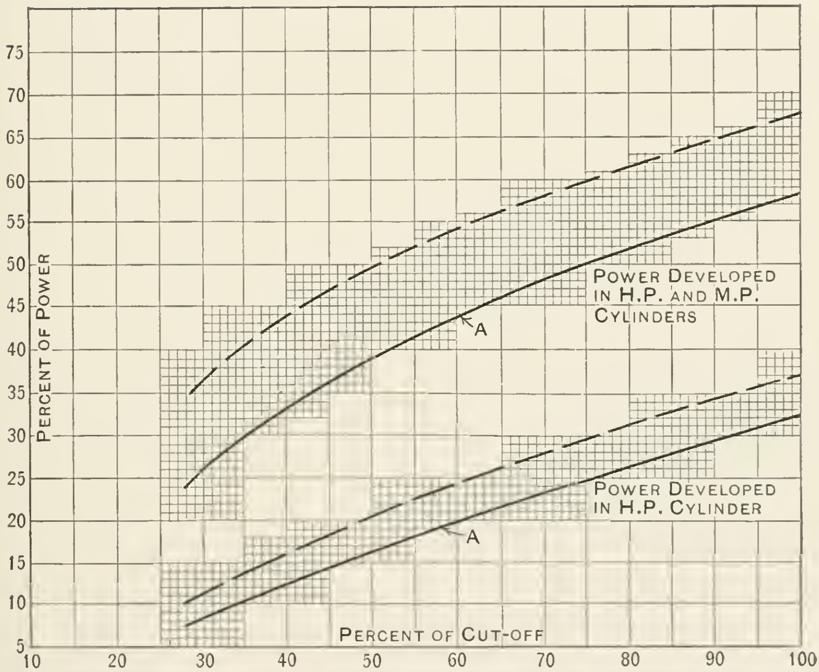


FIG. 13.

The cut-offs for maximum power will be:

M.P. cylinder = 0.60 × 1.4 = 0.84,
L.P. cylinder = 0.605.

The curves of Fig. 13 enable us to determine what portion of the total power will be developed in the different cylinders by these cut-offs or any others that we may care to use.

	M.P. cut-off	L.P. cut-off	Per cent of total work in		
			H.P.	M.P.	L.P.
(1) Max. economy..	0.60	0.605	20	24	56
(2) Max. power....	0.84	0.605	27.5	16.5	56
(3)................	0.76	0.75	25	25	50
(4)................	0.60	0.53	20	20	60

The cut-offs given in line (4) seem to be the best in this case. They will insure that the engine will have its maximum economy, and the engine can be balanced by putting the two L.P. cylinders in the middle of the engine, and the H.P. and M.P. cylinders can be placed at the ends of the engine and the parts made lighter, since each develops less than each L.P. cylinder.

If this engine were to work with a back pressure of 4 pounds absolute instead of 18 pounds absolute the value of R_a would be 8.8, and the cylinder diameters and stroke would be

$$\frac{31'' - 50'' - 57\tfrac{1}{2}''\ (2)}{48''}.$$

The broken lines in Fig. 13 are the curves for this engine and it can be readily seen that the usual distribution of power, namely, H.P.–30 per cent, M.P.–30 per cent, L.P.–40 per cent, could be easily obtained with these proportions.

23. Steam Consumption. — Fig. 14 gives the relation between M.R.P.$_4$ (four pounds back pressure in L.P. cylinder), pounds of steam per I.H.P., and ratio $\frac{P_i}{R_a}$. There seems to be some ground for the statement that the steam per I.H.P. decreases for any given value of $\frac{P_i}{R_a}$ as the M.R.P. is increased.

Engine tests usually determine only the most economical conditions under which that particular engine can be run. The determination of the most economical condition under which power can be generated would necessitate tests where a certain M.R.P. is obtained with different values of P_i and R_a. Fig. 6 shows, for instance, that a M.R.P.$_0$ of 50 pounds can be obtained under the following conditions:

P_i	R_a	$\frac{P_i}{R_a}$	P_i	R_a	$\frac{P_i}{R_a}$
140	5.65	24.7	225	8.40	26.8
160	6.35	25.1	250	9.25	27.0
180	6.95	25.9	275	10.35	26.6
200	7.55	26.5			

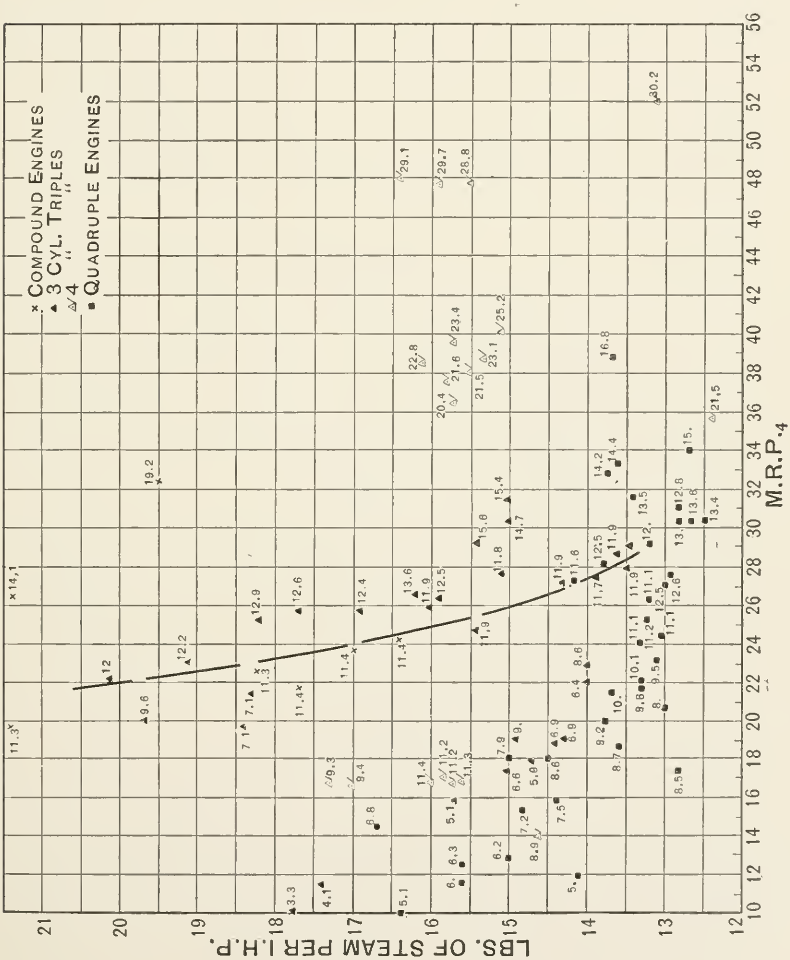


Fig. 14.

It seems probable that some one of these conditions would prove more economical than any of the others. A complete investigation made in this way with different sized engines would determine under what condition power could be most economically generated.

24. Distribution of Work at Reduced Powers. — Fig. 12 gives the distribution of work when the engines are developing full power. As the power is reduced the distribution may change

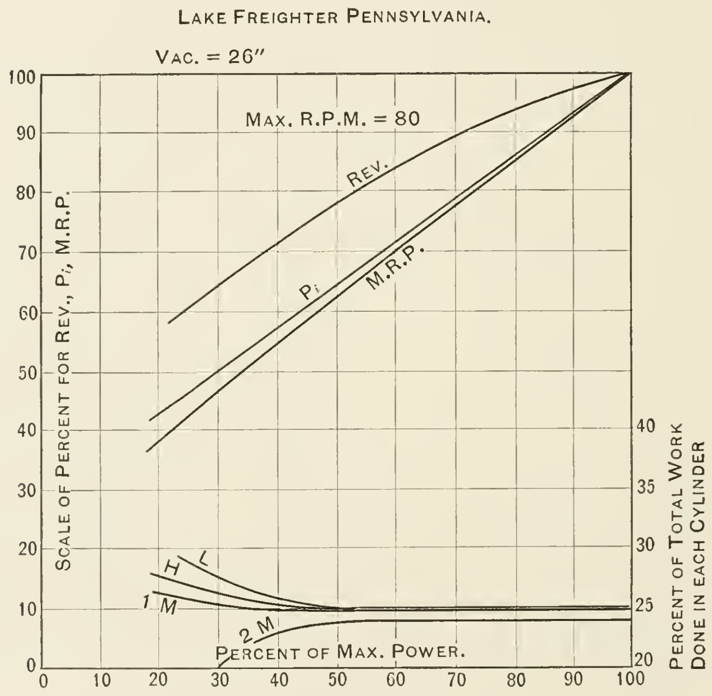


FIG. 15.

somewhat, as shown by Figs. 15 to 18. The results from which these figures were plotted were obtained from engines running with fixed cut-offs, the power being reduced by throttling the steam or reducing the boiler pressure. It will be noticed that in general the percentage of work developed in the L.P. cylinder decreases as the initial pressure falls.

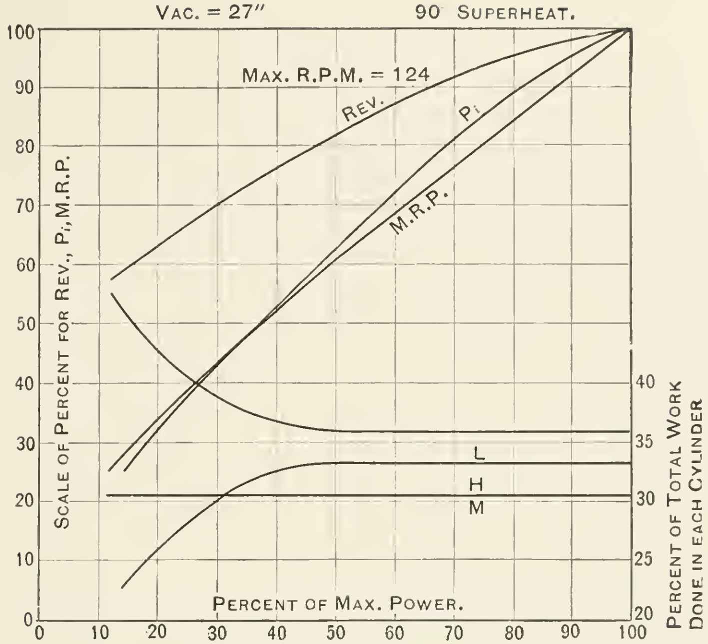


FIG. 16. U. S. S. Michigan.

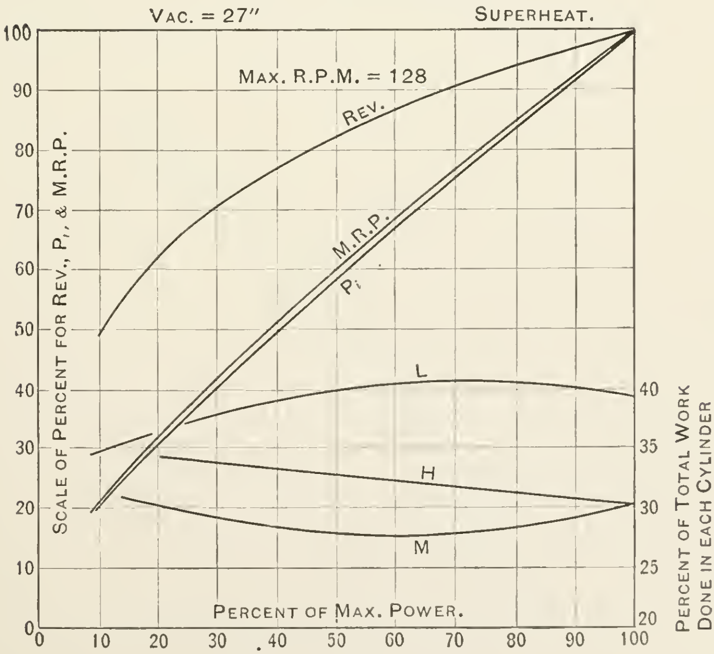


FIG. 17. U. S. S. Delaware.

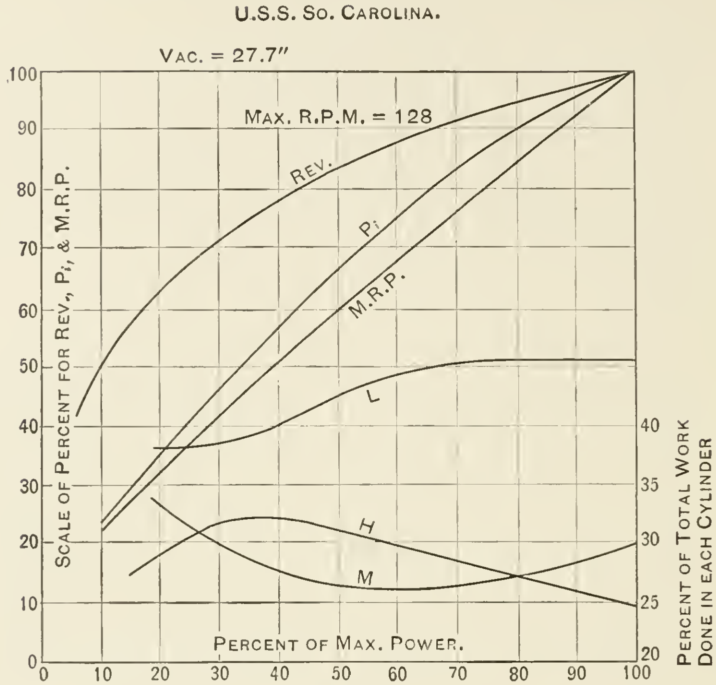

U.S.S. So. Carolina.
Vac. = 27.7″
Max. R.P.M. = 128
Rev.
Pi
M.R.P.
L
H
M
Scale of Percent for Rev., Pi, & M.R.P.
Percent of Max. Power.
Percent of Total Work Done in each Cylinder
100
90
80
70
60
50
40
30
20
10
0
0 10 20 30 40 50 60 70 80 90 100
40
35
30
25
20

Fig. 18.

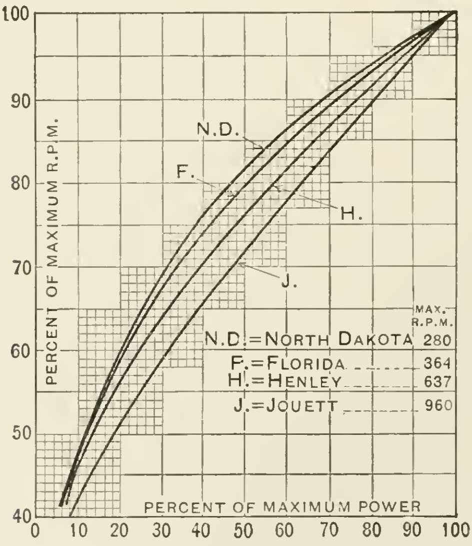

N.D.
F.
H.
J.
Percent of Maximum R.P.M.
Percent of Maximum Power
Max. R.P.M.
N.D.=North Dakota 280
F.=Florida 364
H.=Henley 637
J.=Jouett 960
100
90
80
70
60
50
40
0 10 20 30 40 50 60 70 80 90 100

Fig. 19.

25. Variation of Revolutions and M.R.P. at Reduced Power. — It is interesting to note the manner in which the mean referred pressure and number of revolutions per minute vary as the power is decreased. The revolutions decrease much less rapidly than the mean referred pressure. The greater the maximum number of revolutions, the more rapid the rate of decrease of the revolutions. This is shown by comparing Fig. 15 in which the maximum number of r.p.m. was 80 with Fig. 16 in which the maximum number of r.p.m. was 124. This is also shown more clearly in the case of boats with turbines. In Fig. 19 we have maximum revolutions per minute ranging from 280 to 960, and the rate of decrease of revolution with power is more rapid with the higher revolutions.

In the case of the battle-ship engines shown in Figs. 16 to 18 the mean referred pressure at half power is 60 per cent of that at full power. In other words a decrease of power of 10 per cent is accomplished by a reduction of mean referred pressure of 8 per cent and, approximately, a 2 per cent reduction of revolutions.

SECTION II

DESIGN OF ENGINE PARTS

WORKING STRESS FACTORS

26. Effect of Character of Load. — The working stress factors to be used in the design of the different parts will depend upon the character of the load to which any part is subjected and upon the construction of the part. There are three kinds of load, steady, intermittent, and alternating. The steady load is one which is applied in an appreciable length of time, causing tension or compression of an unchanging amount. The intermittent load is one which is applied more or less suddenly and produces tension or compression of a varying amount. The load may vary from a maximum to zero, or to a minimum other than zero. The smaller the range of variation the more nearly the load approaches a steady load. The alternating load is of such a character as to cause the stress to change alternately from tension to compression.

It has been found by experiment that a piece of metal which has enough sectional area to stand a certain steady load for an indefinite time, will fail after a while if the load is intermittent and varies from a maximum to a minimum, and will fail much sooner if the load is alternating. The relative destructiveness of the three kinds of load is about in the ratio 1, 2, 3. A piece of metal which has enough area to carry a certain steady load indefinitely must have the load reduced to one-half if it becomes intermittent, and reduced to one-third if the load becomes alternating.

27. Working Stress Factors. — Working stress factors are usually based upon the ultimate strength of the material. If the ultimate strength were used as the working stress the part would stand only one application of the load. If the load is to

be applied continually the working stress must not exceed the elastic limit, the limit of the stress to which a material may be subjected and have the strain, or deformation, proportional to the stress.

The elastic limit varies for different grades of steel but for the steel used in marine engines it is about 62 per cent of the ultimate strength. This would require the use of a minimum working stress factor of 1.6 for steady loads, 3.2 for intermittent loads, and 4.8 for alternating loads. It is rarely possible to figure exactly the load to which a part will be subjected and we cannot be sure that the elastic limit will always be 0.62 of the ultimate strength; for these reasons it is usual in land practice to apply a further factor of 2. This gives 3.2, 6.4, and 9.6 as the working stress factors for the three kinds of load.

In marine practice it is usual to increase the factor to 2.5 since the parts are subjected to loads of greater uncertainty, due to the rolling and pitching of the vessel, and also because a breakdown at sea is accompanied by greater danger than one on land. The factors for engines of the merchant marine are thus increased to 4, 8, and 12 for steady, intermittent, and alternating loads.

The factors for naval engines need not be as large as for merchant engines since the latter are worked to nearly their maximum capacity all the time, while naval vessels are ordinarily cruising around at a speed which calls for only 10 or 15 per cent of the maximum power of the engines. In the case of the latter the parts can be designed with lower factors — about 6 for intermittent and 9 for alternating loads. The factor will be very large for cruising powers and yet will not be so small for the short time that the engine works at full power as to reduce the life of the engine to any great extent. Advantage is not taken of this, however, in all classes of naval engines and some of the engines of the larger battleships are designed with as large factors as those of the merchant marine.

In the design of certain parts such as the piston rod, connecting rod, and eccentric rods, allowance must be made for stresses introduced by lack of alignment. As the different pins and

bearings wear the alignment may be destroyed and the loading may become eccentric. In the case of such parts it is usual to increase the working stress factor 50 per cent to allow for these conditions.

There is practically no part of a marine engine which is subjected to a steady load — hence, as low a factor as 4 would never be used. The nearest approach to a steady load is the load condition which exists in the receiver pipes between cylinders, and in the line shafting of balanced engines. In these parts the load is intermittent, varying from a maximum to something more than half the maximum, and a factor as low as 6 may be used. It should be borne in mind that the working stress factors of 8 for intermittent and 12 for alternating loads are used for the purpose of giving the working parts a long life, and that *occasional* loads of such a magnitude as to reduce the factor to 3 can be carried by the parts without injuring the material.

28. Threaded Parts. — Certain parts should be designed with larger factors than those given above, due to the manner of their construction. Bolts and studs are subjected to intermittent loads, but due to the presence of threads the area at the root of the threads is weaker than the same area would be in a

TABLE 5

WORKING LOAD FOR BOLTS

Diam.	Area at root	No. of th'ds	Working load		Diam.	Area at root	No. of th'ds	Working load	
			Merchant	Naval				Merchant	Naval
$\frac{3}{4}$	.302	10	990	1,650	3	5.63	4	33,800	56,300
$\frac{7}{8}$	.419	9	1,410	2,350	$3\frac{1}{4}$	6.73	4	40,400	67,300
1	.55	8	2,060	3,150	$3\frac{1}{2}$	7.9	4	47,400	79,000
$1\frac{1}{8}$	.694	7	2,670	4,050	$3\frac{3}{4}$	9.21	4	55,200	92,100
$1\frac{1}{4}$	.891	7	3,520	5,400	4	10.6	4	63,500	106,000
$1\frac{3}{8}$	1.057	6	4,250	6,500	$4\frac{1}{4}$	12.1	4	72,500	121,000
$1\frac{1}{2}$	1.294	6	5,360	8,300	$4\frac{1}{2}$	13.68	4	82,100	136,800
$1\frac{5}{8}$	1.515	$5\frac{1}{2}$	6,450	10,000	$4\frac{3}{4}$	15.36	4	92,100	153,600
$1\frac{3}{4}$	1.746	5	7,625	11,800	5	17.2	4	103,500	172,000
$1\frac{7}{8}$	2.051	5	9,170	14,400	$5\frac{1}{2}$	21.05	4	126,400	210,500
2	2.302	$4\frac{1}{2}$	10,650	16,800	6	25.3	4	152,000	253,000
$2\frac{1}{4}$	3.023	$4\frac{1}{2}$	14,800	23,500	$6\frac{1}{2}$	30	4	180,000	300,000
$2\frac{1}{2}$	3.719	4	19,400	31,200	7	35	4	210,000	350,000
$2\frac{3}{4}$	4.622	4	25,700	42,000	..	.....	..	....	....

perfectly plain bar. In small bolts, moreover, a considerable initial stress will be present due to the torsion to which they are subjected when the nuts are set up. These two conditions make it necessary to increase the factor to 10 in the case of all bolts 3 inches in diameter and over, and in small bolts the factor should increase as the diameter decreases, becoming about 16 for a bolt 1 inch in diameter. Table 5 gives the loads that various sized bolts can carry when the steel has a tensile strength of 60,000 pounds per square inch.

29. Column Formula. — The sizes of piston rods, connecting rods, and steel columns should be determined by means of a column formula. These parts should all be considered as columns with round or pin ends. Many formulæ for columns are based upon the value of $\frac{l}{r}$, or ratio of length to radius of gyration of cross section. This, however, is not a convenient form for engine work, and for that reason the following formulæ have been devised:

Solid rods or columns:

$$D^2 = \sqrt{\frac{1.8\,FCl^2}{E} + F^2} + F. \qquad (12)$$

Hollow rods or columns:

$$D^2 = \sqrt{\frac{1.8\,FCl^2}{E} + F^2 + (2\,F + d^2)\,d^2} + F. \qquad (13)$$

D = diameter in inches of rod at middle of length.

C = ultimate strength of material in pounds per square inch.

l = length of column in inches.

E = modulus of elasticity of material.

d = internal diameter of hollow column.

$F = \dfrac{2\,Wn}{\pi C}$.

W = *maximum* load upon column in pounds.

n = working stress factor, to be determined by character of load and construction of part.

These formulæ are good for steel, iron, composition, or wooden columns, and the following values of E and C should be used:

Material	E	C
Steel	30,000,000	60,000–95,000
Wrought iron	28,000,000	48,000
Cast steel	30,000,000	65,000–80,000
Cast iron	13,000,000	15,000–20,000 (tension) 100,000 (compression)
Composition	13,000,000	40,000–60,000
Oak	2,000,000	8,500 (compression)
Yellow pine	2,000,000	8,000 (compression)
Oregon pine	1,700,000	5,700 (compression)
Spruce and ash	1,600,000	7,200 (compression)
White pine	1,400,000	5,400 (compression)

30. Hollow Columns. — In naval engines it is often desirable to make all piston and connecting rods of the same outside dimensions, but to bore out the rods which carry the lighter loads and thus get the necessary difference in weights of reciprocating parts to give good balance. In this case the value of F should be found, using the lower value of W for the cylinders developing the lesser horse-power. This value of F and the value of D found for the other cylinders can be introduced in the following equation to get the diameter d for the bore:

$$d^2 = \sqrt{D^4 - 2\,FD^2 - \frac{1.8\,FCl^2}{E} + F^2} - F. \qquad (14)$$

The relation of Formula (12) to others in use is shown in Fig. 20. All the curves are drawn for steel of 60,000 pounds tensile strength, and a working stress factor of 4. It will be seen that Formula (12) gives results agreeing very closely with the Pencoyd tables up to a value of $l \div r$ of 125, and from there on the results agree with the tables used by the Bureau of Construction and Repair of the U. S. Navy.

31. Bearing Pressures. — It is usual to figure the bearing surface of pins and shafts as equal to the length of the surface multiplied by the diameter of the pin or shaft. The pressure allowed must be low enough to keep the bearing cool, and varies

with the conditions and the possibility of artificial cooling. When there is no motion between two parts the pressure can be very large, but as the extent of the motion between them increases, the pressure allowed decreases. Since it is a question of keeping the bearing cool, we can deal with the mean unit bearing pressure, as the maximum will occur generally but for an instant. In some cases, however, it is more convenient to

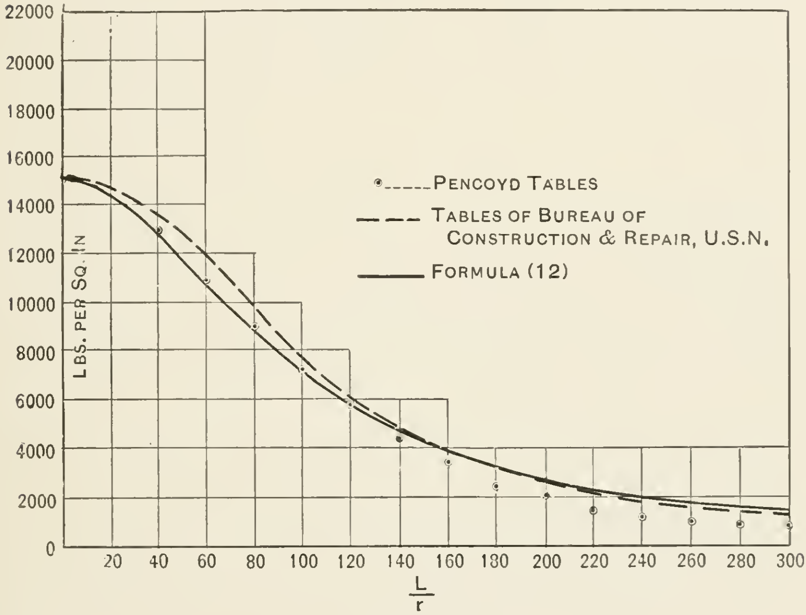


FIG. 20.

deal with the maximum pressure to avoid confusion. The pressures allowed upon the different surfaces are shown in Table 6. All but two of the surfaces are figured for maximum loads, as these loads are used in the design of other parts and it is less confusing to deal with but one load. It will be noticed that those surfaces which have but little relative motion have the greatest unit pressure allowed upon them, while those with more motion have a lower unit pressure. The determination of the loads coming upon the different surfaces will be taken up later.

TABLE 6

Using mean loads	Merchant	Naval
	pounds	pounds
Crank pin	200 to 250	300 to 350
Main bearings:		
1st cylinder	150 to 200	250 to 300
2nd cylinder for'd bearing	200 to 250	300 to 350
2nd cylinder aft. bearing	250 to 300	350 to 400
3rd cylinder for'd bearing	300 to 325	400 to 425
3rd cylinder aft. bearing	325 to 350	450 to 500
4th cylinder for'd bearing	350 to 375	500 to 525
4th cylinder aft. bearing	375 to 400	550 to 600
Using maximum loads		
	pounds	pounds
Slipper guide	50 to 70	80 to 100
Crosshead pins	850 to 1000	1000 to 1200
Link block pin	750 to 1000	850 to 1200
Link block gibs	250 to 400	350 to 500
Eccentric rod pins	700 to 950	900 to 1100
Drag rod pins	500 to 700	700 to 800
Eccentrics	75 to 150	150 to 200
Thrust collars	50 to 80	80 to 100

SHAFTING

32. Types of Shafting. — There are four kinds of shafts used in transmitting the power of the engine to the propeller; the propeller shaft, the line shaft, the thrust shaft, and the crank shaft. The crank shaft is subjected to twisting and bending, the thrust shaft to twisting, the line shaft to twisting, and the propeller shaft to both twisting and bending.

It is usual to figure the size of the crank shaft from the twisting and bending forces acting upon it, and then to base the sizes of the other shafts upon the crank shaft.

There are two types of crank shaft in use for marine engines, the "solid forged" and the "built up." The "solid forged" is used mainly for naval engines and for fast yachts, where weight must be saved. The shafting is usually made hollow and the steel is high grade and oil tempered, in consequence of which the cost is high. The "built up" shaft is heavier, but since it is composed of a number of small forgings it can be made more

cheaply than the solid forged. The "built up" type is common to engines of the merchant marine. There is an intermediate type in which the crank pin and webs are in one forging.

33. Equivalent Twisting or Bending Moments. — The stress developed at the last bearing in the crank shaft by the twisting and bending to which it is subjected can be calculated by finding either the equivalent twisting moment or the equivalent bending moment which will produce a stress equal to that produced by the combined twisting and bending.

If the *equivalent twisting moment* is desired the following formula can be used:

$$T_1 = B + \sqrt{T^2 + B^2}. \tag{15}$$

If the *equivalent bending moment* is desired the following formula can be used:

$$B_1 = 0.35\,B + 0.65\,\sqrt{T^2 + B^2}. \tag{16}$$

B = maximum bending moment acting upon shaft.
T = maximum twisting moment acting upon shaft.

34. Mean Twisting Moment. — The mean twisting moment acting upon the shaft will be

$$t = \frac{\text{I.H.P.} \times 33{,}000 \times 12}{2\,\pi n} \text{ (inch pounds).} \tag{17}$$

I.H.P. = total indicated horse-power of engine.
n = revolutions per minute.

The relation between the mean twisting moment and the maximum can be determined by analyzing the results obtained from engines which have been built. In any such computations the effect of inertia should be considered, as the maxima obtained by disregarding inertia effects will be considerably smaller than if inertia is taken into account, especially in the case of three-crank engines. When the cranks are at 180° or at 90° the inertia effects are nearly balanced. The importance of considering inertia is shown by the following table, which gives results of calculation made upon certain engines:

TABLE 7

	Boston 3 crank low leading	U.S.S. *Monterey* 3 crank high leading	U.S.S. *New York* 3 crank high leading	U.S.S. *Olympia* 3 crank low leading	*Siberia* 4 crank
$\frac{\text{I.H.P.} \times 33{,}000}{2\,\pi r n}$ (1)	72,000	72,000	102,500	181,000	179,000
Max. turning force,— inertia forces included (2)	101,500	109,000	145,000	279,000	245,500
$\frac{(2)}{(1)}$ (3)	1.41	1.51	1.42	1.55	1.37
Max. turning force, — without inertia forces. (4)	84,000	93,500	125,250	233,000	240,000
$\frac{(4)}{(1)}$	1.17	1.3	1.22	1.29	1.34
$\frac{(2)}{(4)}$	1.21	1.16	1.16	1.20	1.02

The maximum forces were, on the average, about 18 per cent larger for three-crank engines when inertia was considered than when it was neglected.

35. Maximum Twisting Moment. — The maximum twisting moment can be obtained from the mean twisting moment by use of certain factors:

$$T = ct = c\,\frac{\text{I.H.P.} \times 33{,}000 \times 12}{2\,\pi n}. \qquad (18)$$

$$\begin{aligned} c &= 2.0, \text{ single-crank engine} \\ &= 1.67, \text{ two-crank engine} \\ &= 1.5, \text{ three-crank engine} \\ &= 1.35, \text{ four-crank engine.} \end{aligned}$$

The crank shaft will be subjected to a maximum bending moment at the last bearing, and an investigation of a number of cases showed that the maximum bending occurred at practically the same time as the maximum twisting. The same forces which are acting to produce the maximum twisting moment are acting to produce bending. These forces are — the thrust of the connecting rod of the last cylinder; the rotative force transmitted through the crank pin from the forward cylinders; the centrifugal force arising from the rotation of the crank webs,

crank pin, and lower part of the connecting rod; the weight of the reciprocating parts of the last cylinder.

36. Maximum Bending Moment. — The force which produces maximum bending was found upon investigation to be about 0.8 of the force producing the maximum twisting moment in the case of three-cylinder engines, and equal to the above force in the case of four-cylinder engines.

It will be assumed that the bending moment upon the crank shaft will be given by the formula:

$$B = \frac{Wl}{8}. \tag{19}$$

$$W = 0.8\frac{T}{r} \text{ (three-cylinder engines)}$$

$$= \frac{T}{r} \text{ (four-cylinder engines).}$$

r = length of crank arm in inches.

l = distance from center to center of bearings of last cylinder

= 0.7 diameter of L.P. cylinder (three-cylinder Triples, and Quadruples)

= 0.85 to 0.95 diameter L.P. cylinder (four-cylinder Triples, merchant)

= 0.8 to 0.85 diameter L.P. cylinder (four-cylinder Triples, naval).

37. Shaft Diameter from Equivalent Twisting Moment. — If the equivalent twisting moment is to be used in finding the stress the formula will be

$$f = \frac{T_1 r}{I_P}.$$

$$r = \text{radius of shaft} = \frac{D}{2}.$$

$$I_P = \frac{\pi D^4}{32} \text{ (solid shafts)}$$

$$= \frac{\pi (D^4 - d^4)}{32} \text{ (hollow shafts).}$$

$$\therefore \ f = \frac{16\,T_1}{\pi D^3}$$

or
$$D = 1.72\sqrt[3]{\frac{T_1}{f}} \text{ (solid shaft).} \tag{20}$$

If the diameter of hole in hollow shaft $= d = cD$,

$$D = 1.72\sqrt[3]{\frac{T_1}{f(1 - c^4)}}. \tag{21}$$

When the bending and twisting moments are taken as above a working stress factor of 8 can be used. A number of shafts whose diameters conformed to Lloyd's rules were analyzed by this method and the stresses varied from 6600 pounds per square inch to 8800 pounds per square inch, with an average value of 7500 pounds per square inch.

In naval engines where steel of 95,000 pounds ultimate strength is used the working stress can be 12,000 pounds per square inch.

38. Shaft Diameter from Equivalent Bending Moment. — If the equivalent bending moment is used the formula for stress will be

$$f = \frac{B_1 r}{I}.$$

$$r = \frac{D}{2}.$$

$$I = \frac{\pi D^4}{64} \text{ (solid shaft)}$$

$$= \frac{\pi (D^4 - d^4)}{64} \text{ (hollow shaft).}$$

$$D = 2.17\sqrt[3]{\frac{B_1}{f}} \text{ (solid shaft)} \tag{22}$$

$$= 2.17\sqrt[3]{\frac{B_1}{f(1 - c^4)}} \text{ (hollow shaft).} \tag{23}$$

$$c = \frac{\text{diameter of hole}}{D}.$$

The same shafts mentioned above when analyzed by this method gave stresses ranging from 7900 pounds per square inch to 10,500 pounds per square inch, with an average of 9000 pounds per square inch. This would mean a working stress factor of about 7.

In naval engines a stress of 14,000 pounds per square inch would be permissible.

39. Coupling Bolts. — The different sections of crank shaft and line shaft are bolted together at the coupling flanges by bolts of such a size that their shearing resistance is equal to the shearing resistance of the shaft.

In the case of the shaft

$$f = \frac{Tr}{I_P} = \frac{16\,T}{\pi D^3}.$$

$$\therefore \quad \frac{T}{f} = \frac{\pi D^3}{16}.$$

In the case of the coupling bolts

$$T = fJn\frac{\pi d^2}{4}.$$

$$\therefore \quad \frac{T}{f} = Jn\frac{\pi d^2}{4}.$$

f = unit shearing stress in shaft and bolts.
T = maximum twisting moment in shafting.
D = diameter of shafting.
d = diameter of coupling bolt at face of coupling.
J = radius of pitch circle of coupling bolts.
= about 0.7 D.
n = number of coupling bolts, usually 6 for three-crank engines and 8 for four-crank engines.

Since the shearing stress f and the maximum twisting moment T are the same for bolts and shaft, the quantity $\frac{T}{f}$ is constant.

$$\therefore \quad \frac{\pi D^3}{16} = \frac{Jn\pi d^2}{4}.$$

$$d = \frac{D}{2}\sqrt{\frac{D}{nJ}} \text{ (solid shaft).} \tag{24}$$

$$d = \frac{D}{2}\sqrt{\frac{D\,(1 - c^4)}{nJ}} \text{ (hollow shaft).} \tag{25}$$

$$c = \frac{\text{inside diameter of shaft.}}{\text{outside diameter of shaft.}}$$

The coupling bolts are usually tapered from end to end at the rate of 1 inch per foot. Where the nut is attached at the smaller end the diameter can be reduced by an amount varying from $\frac{1}{4}$ inch to $\frac{1}{2}$ inch. In the crank shaft these nuts serve merely to keep the bolt in place. In the line shaft the area at the root of the threads of the bolts must be sufficient to take the pull of the propeller when the engine is reversed.

40. Sizes of Crank Shaft Parts. — (See Fig. 21.)

	Built up	Solid forged
Thickness of crank web...	$A = 0.6\,D$ to $0.7\,D$	$0.55\,D$ to $0.65\,D$
Diam. of crank pin........	$B = 1.05\,D$ to $1.1\,D$	$1.05\,D$ to $1.1\,D$
Length of crank pin.......	C = see main bearings	
Thickness of coupling.....	$E = 0.25\,D$ to $0.28\,D$	$0.2\,D$ to $0.22\,D$
Main bearing clearance....	$F = \frac{1}{4}$ to $\frac{3}{8}$ inch	$\frac{1}{4}$ to $\frac{3}{8}$ inch
Coupling clearance........	$G = 2$ to 3 inches	2 to 3 inches
Eccentric pad clearance...	$H = \frac{1}{4}$ to $\frac{1}{2}$ inch	$\frac{1}{4}$ to $\frac{1}{2}$ inch
Eccentric pad diam........	$K = D + \frac{3}{4}$ inch	$D + \frac{3}{4}$ inch
Crank web holes..........	$L = B + \frac{1}{2}$ inch	
Crank web radius.........	$M = 0.88\,L$	
Crank web radius.........	$N = 0.93\,L$	
Crank web width..........	O = ...	$1.05\,B$ to $1.1\,B$
Metal between holes......	$P = 0.45\,L$ at least	

The thickness of the crank webs is sometimes increased from 0.6 D at the first cylinder to 0.7 D at the last.

The diameter of the coupling flange should be about 2 $(J + d)$. (See Formula (24).)

41. Lloyd's Rules for Determining Sizes of Shafts (1915–16). — The diameters of intermediate (line) shafts are to be not less than those given by the following formula:

For compound engines with two cranks at right angles —

Diameter of intermediate shaft in inches

$$= [0.4\,A + 0.006\,D + 0.02\,S]\sqrt[3]{P}.$$

For triple expansion engines with three cranks at equal angles —

Diameter of intermediate shaft in inches

$$= [0.038\,A + 0.009\,B + 0.002\,D + 0.0165\,S]\sqrt[3]{P}.$$

For quadruple expansion engines with four cranks —

Diameter of intermediate shaft in inches

$$= [0.033\,A + 0.01\,B + 0.004\,C + 0.0013\,D + 0.0155\,S]\sqrt[3]{P}.$$

A = diameter of high-pressure cylinder in inches.
B = diameter of first intermediate cylinder in inches.
C = diameter of second intermediate cylinder in inches.
D = diameter of low-pressure cylinder in inches.
S = stroke of pistons in inches.
P = boiler pressure above atmosphere in pounds per square inch.

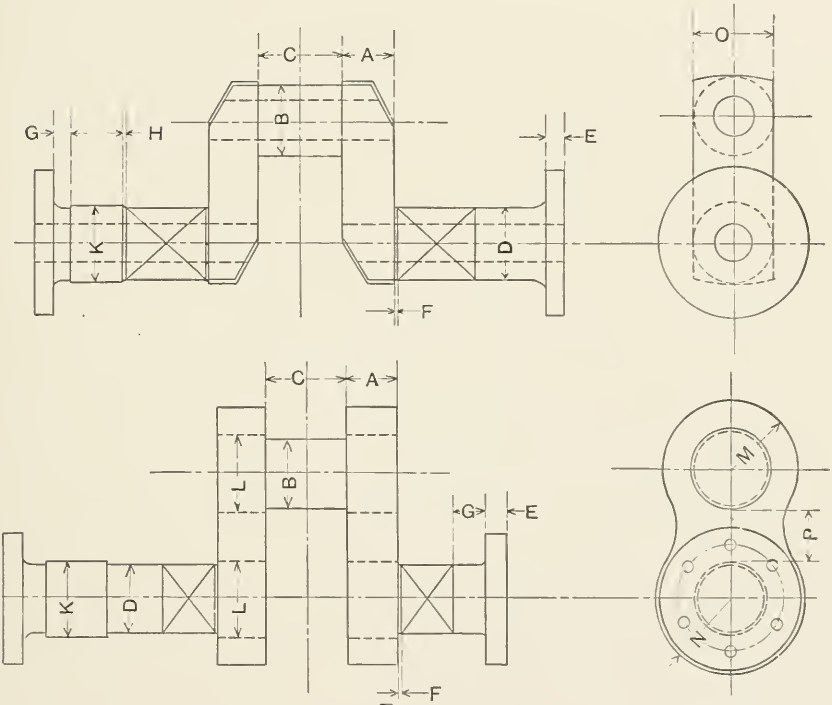


FIG. 21.

The diameter of the crank shaft, and of the thrust shaft under the collars, to be at least $\frac{21}{20}$th of that of the intermediate shaft. The diameter of the thrust shaft may be tapered off at each end to the same size as that of the intermediate shaft.

The diameter of the screw shaft to be equal to the diameter of intermediate shaft (found as above) multiplied by $\left(0.63 + \frac{0.03\,P}{T}\right)$, but in no case to be less than $1.07\,T$, where P is the diameter of

propeller, and T the diameter of intermediate shaft, both in inches. This size of shaft is intended to apply to shafts fitted with continuous liners the whole length of the stern tube, as provided for in Section 8, paragraph 3. If no liners are used or if two separate liners are used, the diameter of the shaft should be $\frac{21}{20}$th that given above.

The diameter of screw shaft is to be tapered off at the forward end to the size of the crank shaft.

42. Internal Combustion Engines. — Rules for Determining Sizes of Shafts. The crank, intermediate, and other shafts if of ordinary mild steel are to be of not less diameters than as given in the following table.

For petrol or paraffin engines for smooth water services:

Diameter of crank shaft in inches $= C\sqrt[3]{D^2S}$,

where D = diameter of cylinder in inches,

S = stroke of piston in inches.

Four-stroke cycle	Two-stroke cycle	Bearing between each crank	Two cranks between the bearings
For 1, 2, 3, or 4 cylinders..	1 or 2 cylinders	$C=0.34$	$C=0.38$
For 6 cylinders...........	3 cylinders	$C=0.36$	$C=0.40$
For 8 cylinders...........	4 cylinders	$C=0.38$	$C=0.425$
For 12 cylinders..........	6 cylinders	$C=0.44$	$C=0.49$

For open sea service add 0.02 to C.

Diameter of intermediate and screw shafts in inches

$$= C\sqrt[3]{D^2S\,(n+3)},$$

where D = diameter of cylinder in inches,

S = stroke of piston in inches,

n = number of cylinders.

For smooth water services	For open sea services
$C=0.155$ for intermediate shafts.....................	$C=0.165$
$C=0.170$ for screw shafts fitted with continuous liners.	$C=0.180$
$C=0.180$ for screw shafts fitted with separate liners or with no liners....................................	$C=0.190$

In engines of two-stroke cycle n is to be taken as twice the number of cylinders.

When ordinary deep thrust collars are used the diameter of the shaft between the collars is to be at least $\frac{21}{20}$th of that of the intermediate shaft. In the cases of *Diesel* and other engines in which very high initial pressures are employed, particulars should be submitted for special consideration.

TORSIONAL VIBRATION OF SHAFTING

When the turning moment exerted upon a shaft varies in intensity there will be a certain amount of torsional vibration of the shafting. This vibration will introduce large stresses if the variation in the intensity of the turning moment coincides with the natural period of vibration of the shafting, or with some multiple of that period.

43. Masses Affecting Torsional Vibration. — In determining the period of vibration of the shafting the masses of the following parts must be used, — the propeller and entrained water, propeller shaft, line shaft, thrust shaft, crank shaft, connecting rod ends, and reciprocating parts. Approximations must be made in certain cases. The mass of the water entrained by the propeller is assumed to be 25 per cent of its mass. The cranks are reduced to one equivalent crank at the middle of the length of the crank shaft. The reciprocating parts travel a distance equal to twice the stroke for every revolution while the crank pin travels a distance of π strokes. For this reason it is usual to assume that the effect of the reciprocating parts is equivalent to that of a single mass at the crank pin equal to $\frac{2}{\pi}$ the mass of the reciprocating parts.

44. Equivalent Masses at Crank Circle. — All of these masses must be reduced to equivalent masses at the crank circle. This can be done by finding the polar moment of inertia of the parts about the center line of the shaft and then placing at a distance r such masses as will give the same moment of inertia.

$$\text{Equivalent masses at crank pin} = I_p \left(\frac{1}{r}\right)^2 \frac{1}{32}.$$

I_p = polar moment of inertia (pounds-feet^2).
r = radius of crank pin circle in feet.

45. Relation between Force and Amplitude of Vibration. —

Let m = mass of a piece of shafting reduced to the crank circle.
K = force at crank circle necessary to produce a twist of 1 inch arc measured at crank circle.

The time for one complete vibration of such a shaft will be

$$t = 2\pi\sqrt{\frac{m}{K}}, \tag{26}$$

or

$$K = \frac{4\pi^2}{t^2} m.$$

Let

$$\omega = \frac{2\pi}{t}.$$

Then $K = \omega^2 m$ for 1 inch arc of amplitude on radius r.
$sK = F = s\omega^2 m$ = force for amplitude s.

46. Angle of Twist. — $G = \dfrac{Tl}{\theta I_p}$.

T = twisting moment (foot pounds).
l = length of shaft (feet).
θ = angle of twist in circular measure.
I_p = polar moment of inertia of shaft (square inches-feet^2).

Then

$$\theta = \frac{Tl}{GI_p} = \frac{Frl}{GI_p}. \tag{27}$$

F = force acting at radius r.

In the case of a shaft twisted by the inertia of its own mass, the amplitude of the arc of torsion s varies as the distance x from the origin.

$$s = cx.$$

$$F = s\omega^2 m = cx\omega^2 m.$$

$$\theta = \frac{cx\omega^2 mrl}{GI_p}. \tag{28}$$

If m_1 = mass of shaft per unit length, the angle between two successive infinitesimal masses dx apart =

$$d\theta = \frac{cx\omega^2 m_1\, dx\, rx}{GI_p} = \frac{c\omega^2 r}{GI_p} m_1 x^2\, dx.$$

The angle at the end of the shaft at a distance l from the origin:

$$\theta_m = \frac{c\omega^2 r}{GI_p} \int_0^l m_1 x^2\, dx = \frac{c\omega^2 r}{GI_p} \frac{l^3 m_1}{3}.$$

Since $\qquad lm_1 = m,$

then $\qquad \theta_m = \frac{c\omega^2 r}{GI_p} \frac{m}{3} l^2.$

From (28) a single mass m_2 at a distance l from the origin would by its own inertia produce an angle of twist,

$$\theta = \frac{c\omega^2 r}{GI_p} m_2 l^2.$$

Therefore we can substitute for the mass of the shaft distributed over a length l, a single mass of one-third the mass of the shaft at a distance l from the origin.

47. Equivalent Shaft Length for Reduced Diameter. — When the shafts are of different diameters they should be reduced to the diameter of the smallest shaft, and the length decreased enough to make the torsional angle, for a given twisting moment, the same for the actual and reduced diameter.

Let
l = length of shaft.
l_1 = reduced length of shaft.
d = diameter of shaft.
d_1 = diameter of smallest shaft.

From Formula (27), θ will remain constant if

$$\frac{l_1}{d_1^4} = \frac{l}{d^4}, \qquad \therefore \quad l_1 = l\left(\frac{d_1}{d}\right)^4. \qquad (29)$$

48. Crank-shaft Mass and Propeller Mass. — The vibrating masses are reduced to two equivalent masses, one at the propeller and one at the middle of the crank shaft. If these two

masses are vibrating freely they must have the same period and the momentum of one mass must be equal but opposite to the momentum of the other. Between the two masses will be a node where the shaft will have no torsional vibration. This node will be at the center of gravity of the system.

Let M_1 = mass at propeller.
M_2 = mass at middle of crank shaft.
L_1 = distance of M_1 from node.
L_2 = distance of M_2 from node.
S_1 = amplitude of vibration of M_1.
S_2 = amplitude of vibration of M_2.

Momentum of $M_1 = M_1S_1 = M_1CL_1$.
Momentum of $M_2 = M_2S_2 = M_2CL_2$.

$$M_1CL_1 = M_2CL_2.$$

$$\therefore \quad \frac{M_1}{M_2} = \frac{L_2}{L_1},$$

or the node is at the center of gravity of the system.

The propeller mass will be made up of the masses of the propeller, entrained water, and the portion $\frac{L_1}{L_1 + L_2}$ of the shafting mass. The crank-shaft mass will be made up of the masses of the cranks, reciprocating parts, and the portion $\frac{L_2}{L_1 + L_2}$ of the shafting mass.

49. Rate of Vibration. — From Formula (27),

$$\theta = \frac{Tl}{GI_p} \quad \text{and } \theta r = \frac{Trl}{GI_p}.$$

If $\quad \theta r = 1'' \quad$ then $T = Kr$

and $$\frac{Kr^2l}{GI_p} = 1.$$

$$\therefore K = \frac{GI_p}{r^2l}.$$

From Formula (26), $t = 2\pi\sqrt{\frac{mr^2l}{GI_p}} = 2\pi r\sqrt{\frac{ml}{GI_p}}.$

The number of oscillations per minute will be

$$n = \frac{60}{t} = \frac{30}{\pi r}\sqrt{\frac{GI_p}{ml}}. \tag{30}$$

r = radius of crank circle in feet.
G = 12,000,000 for ordinary steel.
I_p = polar moment of inertia (square inches-feet^2).
m = mass of vibrating parts.
l = distance from vibrating mass to node (feet).

Since $M_1L_1 = M_2L_2$ we can use either the propeller mass and its distance from the node, or the crank-shaft mass and its distance. n is called the critical number of revolutions for that engine and it is desirable that n should not equal the normal r.p.m. of the engine or any multiple of the r.p.m.

PISTON RODS

50. Load upon Piston Rod. — The piston rod is a cylindrical column with the ends shaped as in Plates 1, 2, and 3. The middle portion carries an alternating load but the ends are so constructed, with shoulders and tapered parts, that the threaded part carries only an intermittent load. The diameter of the middle portion of the rod is calculated by means of the Column Formula (12), while the ends are figured for tension only. The rod should be figured for the maximum load to which it will be subjected, and this load will be

$$W = \frac{2 \times \text{I.H.P.} \times 33{,}000}{\text{P.S.}}. \tag{31}$$

I.H.P. = the maximum indicated horse-power in a *single* cylinder.
P.S. = piston speed of engine in feet per minute.

51. Diameter of Piston Rod. — The piston rod should be treated as a column with free ends whose length is equal to the distance from the under side of the piston to the center line of the crosshead pins. This length will be approximately

$$l = S + Hd + 6'', \text{ merchant engines}$$
$$= S + Hd + 3'', \text{ naval engines.}$$

S = stroke of engine in inches.
Hd = diameter of H.P. cylinder in inches.

If the cylinder bottom is deeper than 1.4 × diameter of the piston rod, the assumed length should be increased. The diameter of column given by Formula (12) should be increased by $\frac{1}{4}$ inch to allow for turning the rod down when it wears unevenly. The working stress factor to be used in the formula should be 18 for merchant engines but can be reduced to 12 for torpedo boat engines. Hollow piston rods are used upon naval engines and some yacht engines where it is desirable to save weight. The diameter of the hole in the rod is usually from 40 to 65 per cent of the outside diameter of the rod. The ultimate strength of the steel used for the rods of merchant engines varies from 60,000 pounds to 80,000 pounds per square inch, while for naval engines it is often as high as 95,000 pounds.

52. Piston-rod Ends. — The shoulders at the ends of the rod can be negative as shown on Plate 3, or positive, see Plate 2. Negative shoulders are usually made not less than $\frac{1}{8}$ inch in width, while positive shoulders are seldom less than $\frac{1}{4}$ inch wide. The taper should be from $2\frac{1}{2}$ inches per foot to 3 inches per foot; if it is made much finer than that it will be difficult to remove the parts. In some cases the taper is omitted, see Plate 1. If the threaded end is 3 inches or more in diameter the area at the root of the thread can be calculated to carry the load with a working stress factor of 10. The number of threads per inch is usually 4, therefore the diameter at the root of the thread must be increased by $1.299 \div 4 = 0.325$ inch to allow for cutting the threads.

The approximate diameter of the rod can be found from the curves in Fig. 22 by adding $\frac{1}{4}$ inch to the diameters there given for different relations of load and length. The curves are constructed for steel of 80,000 pounds ultimate strength and for a working stress of 4450 pounds. For steel of any other ultimate strength the diameter will vary roughly as the fourth root of the ratio of that ultimate strength to 80,000 pounds. When hollow rods are used the diameter will have to be increased about 10

per cent when the diameter of the hole is about 0.6 the diameter of the rod.

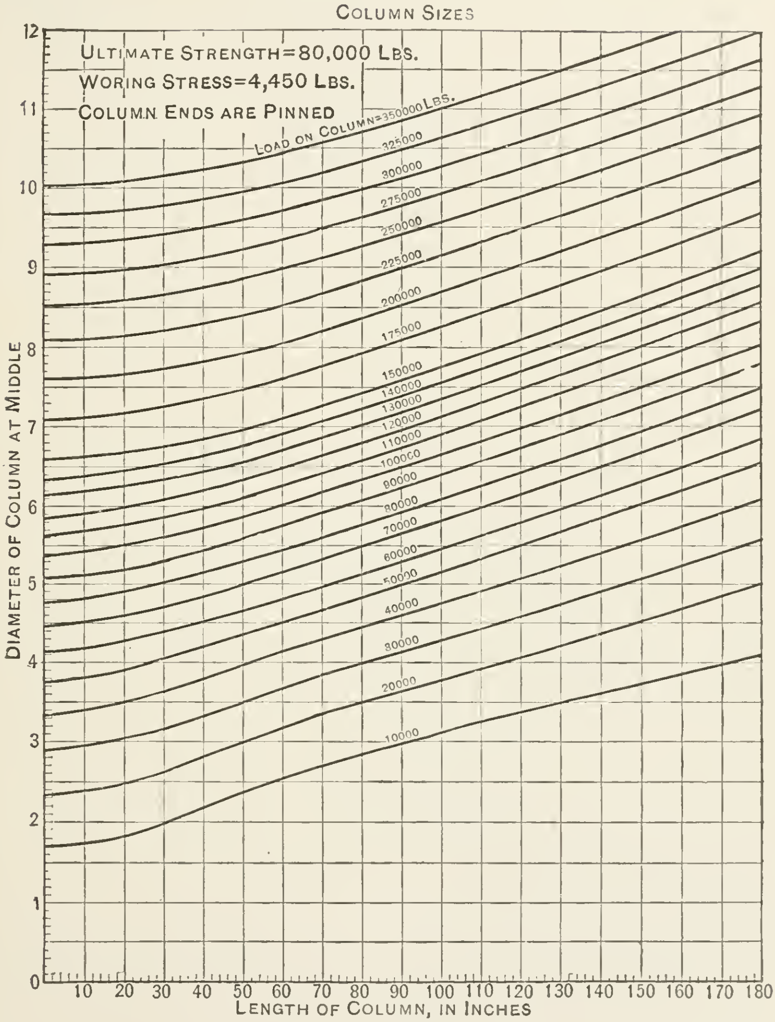


Fig. 22.

CROSSHEADS AND SLIPPERS

53. Types of Crosshead. — It is the function of the crosshead to make the connection between the piston rod and the connecting rod. The connection can be made in three ways: (1) by

means of a block with two pins which bear in boxes in the connecting rod fork, see Fig. 23; (2) by means of a single pin held by the crosshead jaws bearing in a box in the connecting rod, see

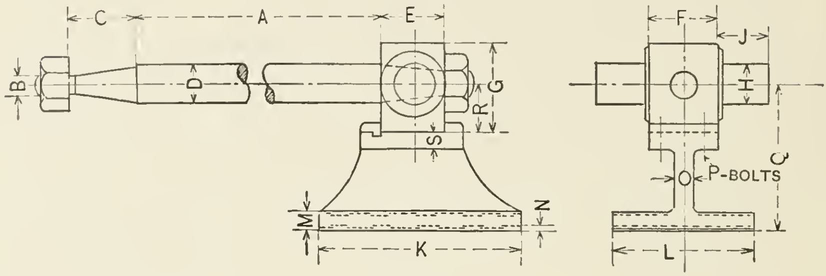


FIG. 23.

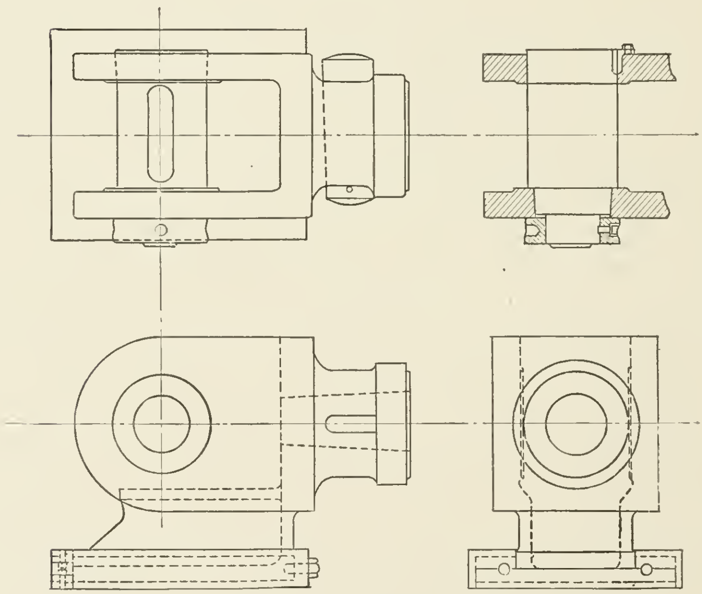


FIG. 24.

Fig. 24; (3) by means of a box on the end of the piston rod which provides bearing for a pin held by the fork of the connecting rod, see Fig. 25. The first is the most common type and is found in both merchant engines and naval engines; the second

type is used on boats running on the Great Lakes; and the third type is used mainly in light high-speed engines such as those for yachts.

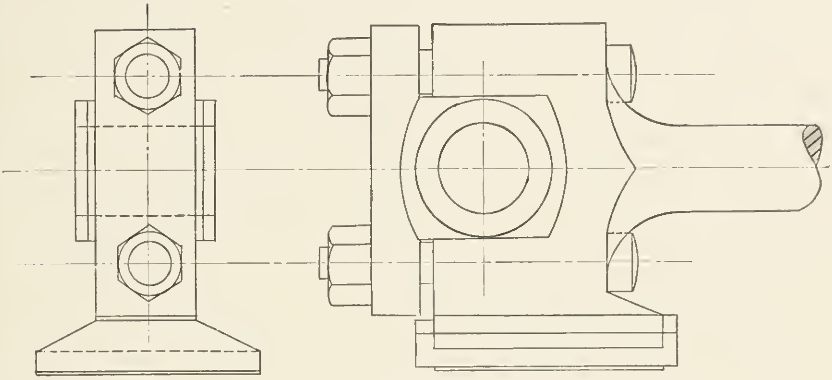


FIG. 25.

54. Size of Crosshead Pins. — The size of the crosshead pin is determined by the amount of bearing surface needed to keep the unit bearing pressure between 850 pounds and 1200 pounds for merchant engines and between 1200 pounds and 1800 pounds for naval engines. The bearing surface is taken as the projected surface of the pin and is equal to the length × diameter. The diameter of the pin or pins is usually from 1.15 D to 1.25 D, and the length is such as to give the necessary surface. The pin is usually made from $\frac{1}{8}$ to $\frac{1}{4}$ inch longer than necessary for surface alone to allow for clearances. (For D, see Fig. 23.)

55. Size of Crosshead Block. — The crosshead block with projecting pins must be figured for splitting on a plane through the center line of the piston rod. It is usual to figure the block as though it were a beam supported at the mid-length of the pins and loaded at the middle of the block with a load equal to W, the piston rod load. The block is approximately a cube and the breadth and depth can be assumed and only the height needs to be figured. The breadth F, see Fig. 26, will be from 1.5 D to 1.7 D. The block is made of forged steel or cast steel and since the load is alternating a working stress factor of 12 must be used.

If we treat the block and pins as a beam supported at mid-length of the pins the length of the beam will be

$$l = F + J$$

and the height of the block, E, can be found from a formula derived from the ordinary beam formula:

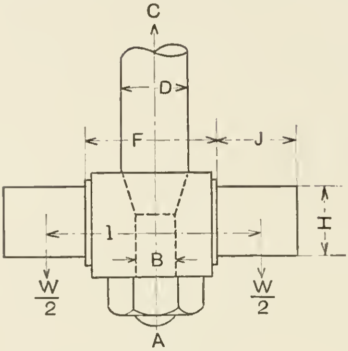


FIG. 26.

$$E = \sqrt{\frac{3\,Wl}{2\,bf}}. \qquad (32)$$

b = net breadth of section = F − mean diameter of hole through block.
f = working stress.
W = maximum load.
$l = F + J$ (see Fig. 26).

Let the depth of the block, G, equal the height, E.

56. Types of Slippers. — The slippers attached to the cross-head blocks are of four different types: (1) the box slipper, see Fig. 27; (2) the single slipper, see Fig. 23; (3) the double

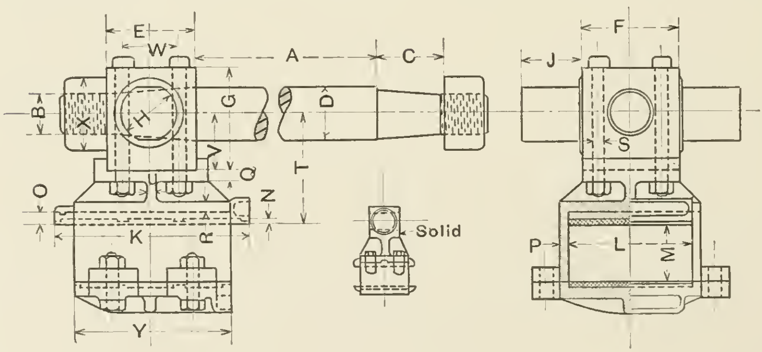


FIG. 27.

slipper, see Fig. 28; (4) the four-slipper type, see Fig. 29. The box slipper is used mainly upon engines of 1000 I.H.P. or less, and also upon engines which are run "backing" as much as "ahead," such as ferry-boat engines. The single slipper is the

most used and is found upon engines of all sizes, both merchant and naval. The double slipper is not used to any great extent since it requires heavy columns upon the front of the engine as well as upon the back to carry the backing guides. The four-

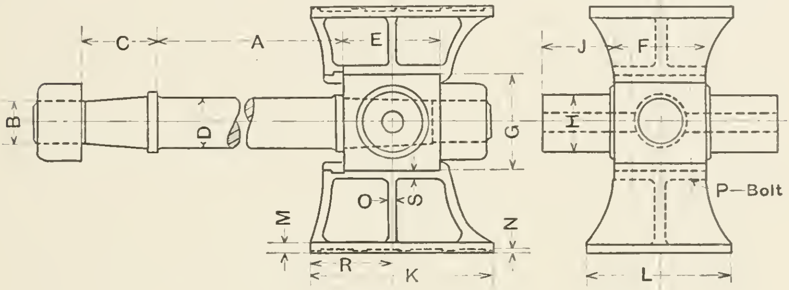


FIG. 28.

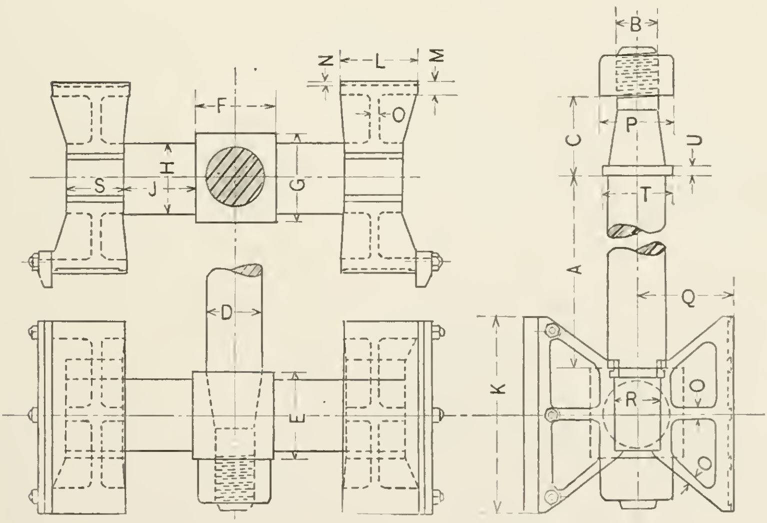


FIG. 29.

slipper type is used upon large engines where the great weight of the cylinders makes it advisable to use four supports rather than two. These four supports can be made to carry the four guide surfaces.

57. Size of Slipper. — The load upon the slipper is zero at the beginning and end of the stroke and a maximum at mid-stroke. This maximum load will be

$$G' = W \tan \sin^{-1}\frac{r}{l},$$

but it is usual to assume it to be

$$G = W\frac{r}{l}. \tag{33}$$

The surface of the slipper should be such that the pressure per square inch is from 55 to 65 pounds for merchant engines, and from 75 to 100 pounds for naval engines. In all types except the single-slipper type the "backing" surface is equal to the "ahead" surface. The breadth of the slipper is usually from 2.5 to 3 times the diameter of the piston rod. In the single-slipper type the connection between the block and the slipper is made by means of a web which must be thick enough to be rigid. The thickness of this web is usually about 0.5 the diameter of the piston rod, and the minimum breadth of the space on the back of the slipper occupied by the web and its fillets will be about 3 inches. In order that the backing surface shall not be less than 75 per cent of the ahead surface the minimum breadth of the single slipper will be 12 inches. The ratio between the length and breadth of the slipper will vary from 1 in small engines to 2 in large engines.

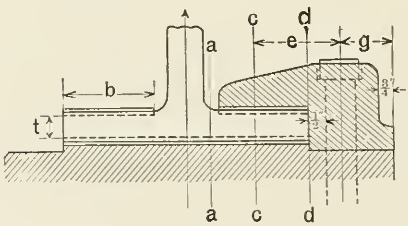


Fig. 30.

58. Thickness of Slipper. — The ahead and backing surfaces are usually lined with white metal of a total thickness of $\frac{1}{2}$ inch. The white metal is held in dovetailed grooves about $\frac{1}{4}$ inch deep so that the surface of the white metal projects $\frac{1}{4}$ inch beyond the body of the slipper and that amount of wear can take place before the steel of the slipper will touch the guide surface. The slipper is usually made separate from the crosshead block and

is attached to it by means of four or more bolts which must be large enough in the case of the box and single slipper to carry the load when backing. The slipper can be made of cast steel or forged steel, and in the case of the single-slipper type should be figured for bending at *aa*, see Fig. 30. It is assumed that the load is acting at the middle of the breadth of the backing surface so that the bending moment coming upon metal is $\frac{G}{2} \times \frac{b}{2}$. The thickness of metal between the backs of the white metal in the single-slipper type should be

$$t = \sqrt{\frac{3\,Gb}{2\,Kf}}. \tag{34}$$

K = length of slipper.
f should be 5000 for forged steel (merchant engines).
8000 for forged steel (naval engines).
3500 for cast steel.

The total thickness of the slipper including white metal will be t + one inch.

59. Backing Guide. — The backing guide can be figured by formula (34) if we assume that only that portion of the guide in contact with the slipper resists the backing load. The only change that will have to be made will be in the value of f. The backing guide is almost always made of cast iron since cast iron and white metal work well together. The value of f should be 1500 for the backing guide and the thickness obtained from the formula will be the thickness at *dd*. The guide can be tapered down to a thickness of 1 inch or $\frac{3}{4}$ inch at the inner end.

60. Backing-guide Bolts. — The backing guide bolts will carry a load greater than G, due to the leverage which the guides exert upon those bolts. The load upon the bolts of one guide will be

$$H = \frac{G\,(e + g)}{2\,g}. \tag{35}$$

e and g have the values shown in Fig. 30. The clearance between the body of the bolt and the edge of the slipper should

be about $\frac{1}{2}$ inch and the thickness of metal outside of the bolts should be about equal to the diameter of the bolts, Q. The lip usually projects about $\frac{3}{4}$ inch. The value of e and g will be as follows:

$$e = \frac{b}{2} + \frac{1''}{2} + \frac{Q}{2}.$$

$$g = 1.5\,Q + \frac{3''}{4}.$$

Assume for the first trial that $\frac{e+g}{g} = 2.25$ and determine the size of bolts upon the assumption that three bolts will carry the load upon one guide. If the diameter so determined gives a value of $\frac{e+g}{g}$ equal to 2.25 approximately, that diameter can be used; if the value is considerably greater, or less, than 2.25, a larger or smaller diameter must be used. The bolts are usually spaced about 6 diameters apart and have cylindrical heads flush with the outside of the guide.

61. Attachment of Slipper. — In the case of the two-slipper and four-slipper types the slippers can be attached to the cross-heads with smaller bolts than in the case of the single-slipper type, since in the former cases the bolts do not have to carry the load when backing. These bolts are usually made from 1 to $1\frac{1}{4}$ inches in diameter. It will be noticed in these two cases that a much stiffer connection between the slipper and block is possible and that the connecting web need be only 1 inch thick.

Removable keys are usually provided on the upper side of the slippers so that they can be taken off without disturbing the alignment of the rod.

CONNECTING RODS

62. Types of Connecting Rods. — Connecting rods are of two general types, the forked marine type and the straight land type. The latter type of rod, see Fig. 31, is used upon some ships on

the Great Lakes but the forked type is the most common elsewhere. The forked type of rod can be divided into two classes: (1) in which the fork carries boxes for the crosshead pins, see Fig. 32; (2) in which the fork carries the pin, see Fig. 33. The latter is used sometimes in naval engines and yacht engines where weight is to be saved.

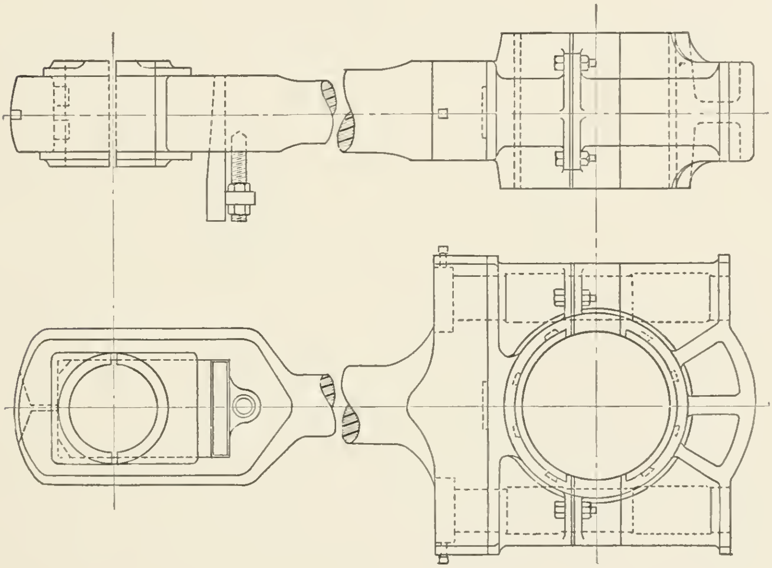


FIG. 31.

63. Diameter of Connecting Rod. — The connecting rod should be figured as a column and Formula (12) can be used for finding the diameter, H, at the middle of its length. The length of the connecting rod is usually between $4\,r$ and $5\,r$; i.e., $\frac{l}{r} = 4$ to 5. The usual ratio is 4.5 but in naval engines where height is of importance a ratio of 4 is used.

The connecting rod and the piston rod are usually made of the same quality of steel, and the same working stress factor should be used, namely 18. The maximum load to which the rod is subjected is somewhat larger than that for the piston rod, due to the fact that the connecting rod makes an angle with the line

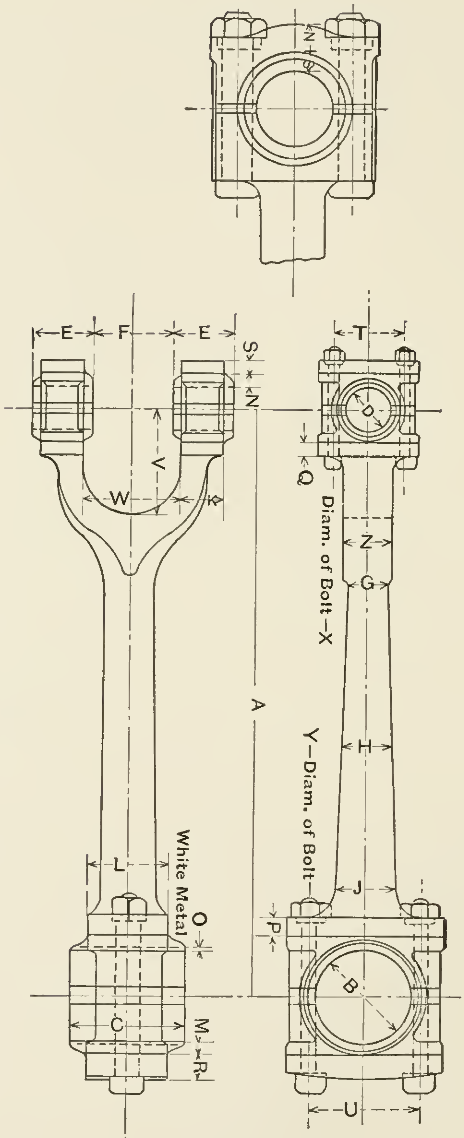

E
F
E
S
N
V
W
K
A
L
White Metal
O
C
M
R
T
D
Q
Diam. of Bolt—X
Z
G
H
Y—Diam. of Bolt
J
P
B
U

FIG. 32.

of centers when the maximum load occurs. The effect of this angularity is to increase the load W to the following extent:

$$P = Wk.$$

If $\frac{l}{r} =$	4	4.25	4.5	4.75	5
$k =$	1.033	1.03	1.026	1.023	1.021

W = maximum load upon piston rod.

The diameter of the rod at mid-length can be determined approximately from the curves of Fig. 22.

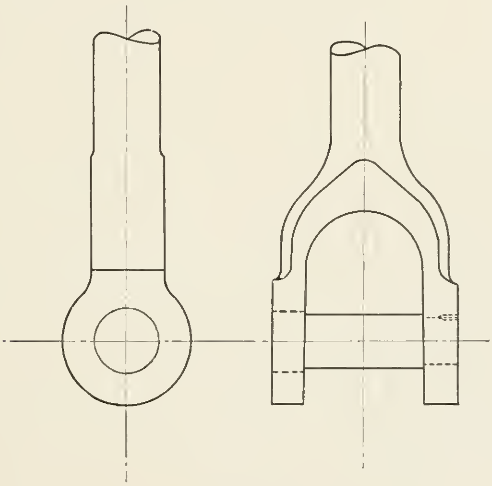


FIG. 33.

64. Taper of Body of Rod. — The accelerating forces, which are acting upon the connecting rod as it swings back and forth across the line of centers, give rise to a certain amount of bending. This bending is a maximum when the rod makes its maximum angle with the line of centers. The resultant of the accelerating forces is acting at the center of percussion, which, in a marine connecting rod, is very close to the crank pin. This subjects the lower part of the rod to bending. There is also a small amount of bending introduced at this point by the friction of the crank pin in the box. The additional stress in the lower part of the rod due to this bending is seldom figured but the diameter of the rod at the lower end is arbitrarily increased. The upper

end of the rod can be decreased in diameter, since the added stress due to column action is less than at mid-length and very little of the cross-bending is felt here. It is usual to increase the diameter at the lower end of the rod to 1.1 H and to make the diameter at the upper end 0.9 H, with a uniform taper from one point to the other.

65. Connecting-rod Bolts. — In a rod of the type shown in Fig. 32 the bolts at either end have to carry the maximum load of the rod. At the crosshead end there are four bolts while at the crank-pin end there are two. The sizes can be determined from the table of bolts, see Table 5. It is safer to assume that the load at the crosshead end will be carried by only three bolts. These bolts are made with cylindrical heads.

66. Connecting-rod Boxes. — The diameter and length of the boxes at the crosshead and at the crank pin must be made to suit the pins working in them. The lugs on the connecting rod to which these boxes are attached have a breadth which is less than the length of the boxes, so that the boxes have a certain amount of overhang. The breadth of the lugs, see Fig. 32, is 0.7 of the length of the box at the crank-pin, and about 0.8 at the crosshead end. The length of the lugs must be sufficient to cause them to project far enough to accommodate the heads and nuts of the bolts. These heads and nuts can cut into the fillets but must clear the body of the rod. The thickness of the lugs must be slightly greater than the diameter of the bolts passing through them. The clearance between the body of the bolts and the pins should be about $\frac{1}{4}$ inch at the crosshead end and about $\frac{1}{2}$ inch at the crank-pin end. The distance between center lines of bolts at the crosshead end must be such that the body of the bolts clear the crosshead pins by the proper amount and the heads of the bolts clear the fork.

67. Connecting-rod Fork. — The thickness, Z, of the fork of the rod, Fig. 32, must be at least equal to the diameter of the rod at the upper end and is often made from $\frac{1}{8}$ to $\frac{1}{4}$ inch thicker, if it does not make the clearance between the body of the bolts and the crosshead pins greater than is desirable. In figuring the fork it is safer to assume the minimum thickness. While

the thickness of the fork is kept constant the breadth of any section must be determined from the bending and direct compression to which it is subjected. The total stress upon any section will be the resultant of the stresses due to direct compression, shear, and bending. We can take these into account by means of the approximate formula

$$f = \frac{P}{2\,bh} + \frac{6\,Pl}{2\,bh^2},$$

$$\therefore h = \frac{P}{4\,bf}\left(1 + \sqrt{1 + \frac{48\,bfl}{P}}\right). \quad (36)$$

h = breadth of fork section corresponding to distance l.

P = maximum load on connecting rod.

b = thickness of fork.

l = distance from neutral axis of section to line of action of force.

f = stress. Use working stress factor of 12.

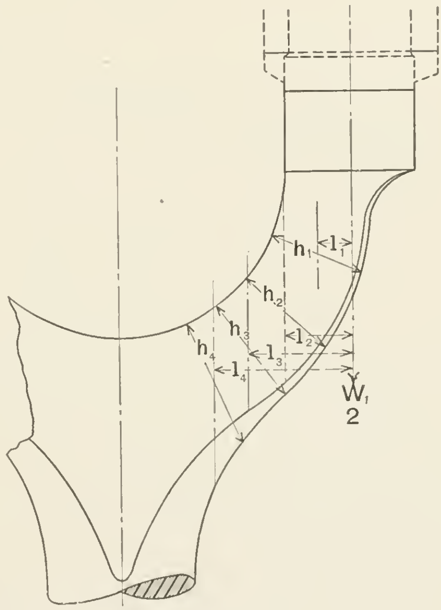


Fig. 34.

Let l have successive values such as 1 inch, 2 inches, 3 inches, etc., as shown in Fig. 34, and find the values of h. The contour of the inside of the fork is made semicircular and the values of h are to be laid off about normal to the semicircle so that the center of the breadth h is the proper distance from the line of action of the force. The outside contour of the fork should consist of arcs of circles, for convenience in laying off. Usually two arcs of the same radius can be made to coincide closely with the points. Forks are sometimes made with the outside

contour a straight line; this makes the fork heavier as the straight line must include the extreme point. The lowest point of the inside of the fork should clear the end of the piston rod by about the thickness of the nut on the end of the rod. A working stress factor of 12 should be used.

68. Connecting-rod Caps. — The caps of the boxes should be figured as beams with a length equal to the distance between bolt centers and a breadth equal to the breadth of the lugs. The conditions of loading and support are such that the beam does not fall into any one of the usual classes; the load is partially distributed and the condition of the ends is somewhere between that of being fixed and being supported. A bending moment of $\frac{Wl}{6}$ will be a close enough approximation. Since the section of the beam is rectangular the height of it will be given by the following formula:

$$\text{Crosshead end } S = \sqrt{\frac{Pl}{2\,bf}}. \qquad (37)$$

$$\text{Crank-pin end } S = \sqrt{\frac{Pl}{bf}}. \qquad (38)$$

P = maximum load in the connecting rod.
l = distance between bolt centers.
b = breadth of cap.
f = allowable stress.

The load upon the caps is intermittent and the lowest working stress factor would be 8, but for the sake of stiffness it should be 10. The caps are made of bronze, cast steel, and wrought steel, the latter being the most common. When the cap and brass are all in one piece the cap should have the required thickness exclusive of the white metal.

69. Connecting-rod Brasses. — The brasses are shells carrying the white metal for the pins and enclosing the bolts. The thickness of metal around the bolts is about one-fourth the diameter of the bolt, and the total thickness of the white metal is from $\frac{1}{2}$ to $\frac{3}{4}$ inch. The two brasses are separated by a thick cast iron liner of horseshoe shape, from 1 to $1\frac{3}{4}$ inches thick, and

also by several tin liners whose thickness varies from $\frac{1}{64}$ to 1 inch. The least thickness of the brasses should be about 0.2 × diameter of pin. The distance from the center of the pin to the back of the brasses should be two-thirds the diameter of the pin.

PISTONS

70. Types of Pistons. — There are two types of pistons in general use, the conical, cast-steel type, see Fig. 35, and the hol-

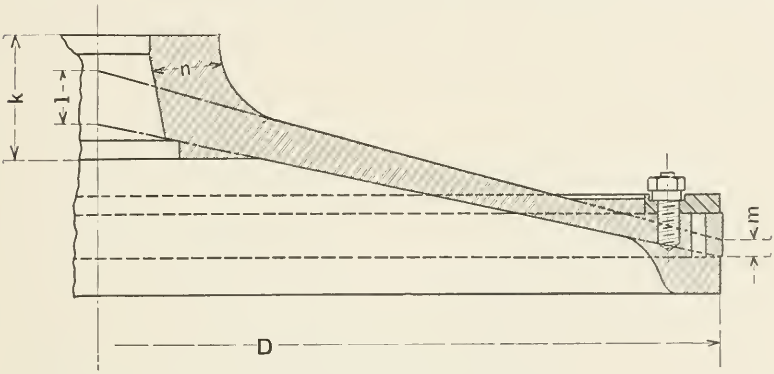


Fig. 35

low, cast-iron, box type, see Plates 1, 2, and 3. The cast-steel piston is lighter and stiffer than the cast-iron, and is used where weight must be saved, and where the engine makes a high number of revolutions per minute. The cast-iron piston gives a cylinder of simpler construction, but the piston cannot be run much over 100 r.p.m.

71. Cast-iron Piston.

Depth of piston at center = 1.5 d.
Depth of piston at rim = 1.2 d.
Thickness of face and back of piston = 1 inch.
Thickness of metal around piston rod = 0.5 d.
d = diameter of piston rod.
Pitch of follower studs = 6 diameters on H.P.
= 7 diameters on M.P.
= 8 diameters on L.P.
Thickness of ribs of piston = $\frac{7}{8}$ inch.

72. Cast Steel Pistons.

Depth of boss at center $= k = d$ (naval engines).

Depth of boss at center $= k = 1.2\,d$ to $1.4\,d$ (merchant engines).

Thickness of metal around piston rod $= n = 0.4\,d$.

Thickness of piston metal at center $= l = \dfrac{D}{200}\sqrt{P} + 0.5$ inch.

Thickness of piston metal at rim $= m = 0.4\,l$.

d = diameter of piston rod.

D = diameter of cylinder.

P may be taken as follows:

Triple	Quadruple
H.P. cylinder, 0.5 boiler pressure	0.45 boiler pressure
1 M.P. cylinder, 0.25 boiler pressure	0.20 boiler pressure
2 M.P. cylinder	0.175 boiler pressure
L.P. cylinder, 0.20 boiler pressure	0.10 boiler pressure

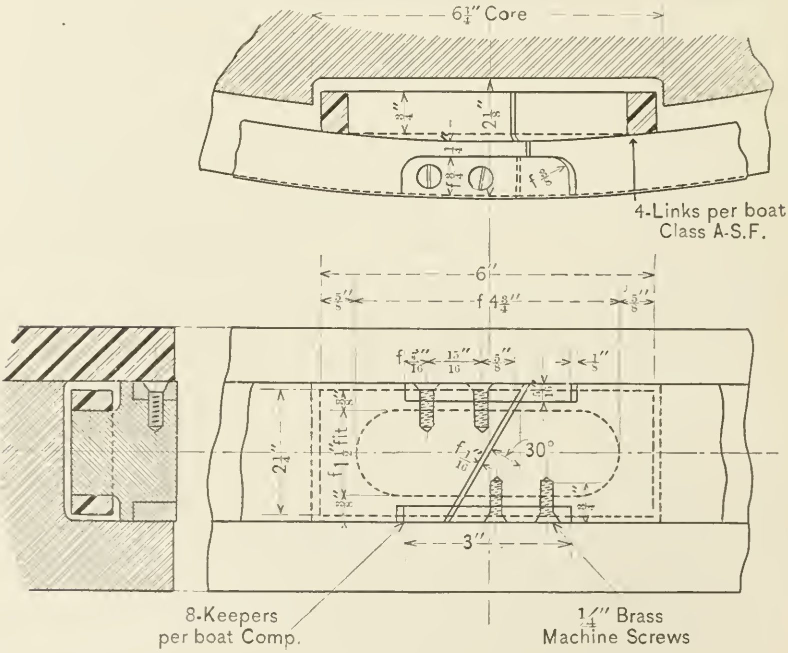
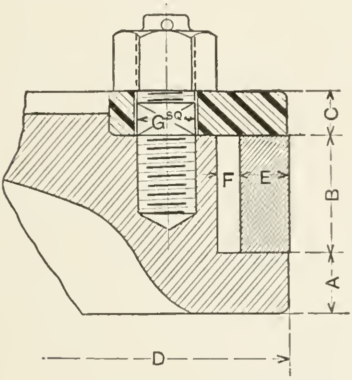


FIG. 36.

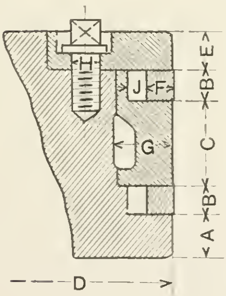
The over-all height of cast-steel pistons is usually made the same for all cylinders, and the details of the boss and of the rims are usually the same for all. The pistons will have a different slope in each cylinder, but in the L.P. this slope on the under side should not be less than 1 in 6 for engines using steam of 175 pounds pressure, and not less than 1 in 3 if the steam pressure is 300 pounds.

73. Piston Rings. — The piston rings are made of cast iron and are usually of the restrained type, as shown in Fig. 36. The



D	A	B	C	E	F	G
36"	2"	2"	1⅛"	¾"	½"	1⅛"
48"	2⅛"	2¼"	1¼"	⅞"	⅝"	1⅛"
60"	2¼"	2⅜"	1⅜"	1"	⅞"	1¼"
OVER 70"	2⅜"	2½"	1½"	1"	1"	1¼"

Fig. 37.



D	A	B	C	E	F	G	H	J
48"	1½"	1⅛"	3"	1½"	1"	2¼"	1⅛"	¾"
60"	1½"	1⅛"	3"	1½"	1"	2¼"	1⅛"	¾"
72"	1⅝"	1⅛"	3¼"	1⅝"	1⅛"	2⅜"	1¼"	¾"
84"	1⅝"	1¼"	3¼"	1⅝"	1⅛"	2⅜"	1¼"	¾"
96"	1¾"	1¼"	3½"	1¾"	1⅛"	2½"	1¼"	⅞"

Fig. 38.

rings are cast solid and larger in diameter than the cylinder they are to fit. A lug is cast upon the inside of the ring and at this place a piece of the ring is cut out. The ring is then sprung together, held by a strap or bolt at the lug, and turned to fit the cylinder. As the ring wears it is let out enough to take up the

wear but not enough to press unduly upon the liner or barrel. With unrestrained rings the steam works in back of the rings and forces them out upon the liner causing a large loss in friction.

74. Piston Rims. — Some details of piston rims are shown in Figs. 37 and 38, and the proper proportions for certain sizes of L.P. cylinders are given in the accompanying tables.

CYLINDERS AND COVERS

75. Cylinder Castings. — Cylinders are usually cast with the barrel, bottom, and valve chest, in one piece. Occasionally the valve chest is a separate casting bolted to the cylinder, and in the case of some torpedo boat engines the cylinder bottom is bolted on, but this latter construction is unusual. The cylinder casting is quite complicated and for this reason it is desirable to use a soft grade of cast iron which will run easily and make a good casting. A soft grade of cast iron, however, does not develop a good wearing surface for the piston rings, and for this reason liners made of a hard-grained iron are often used. If a cylinder is to be jacketed with steam there must be a liner, but, for the reason given above, liners are often used when no jacket steam is to be employed. Even if the space between the liner and the barrel is not to be used for jacketing it is generally connected to the steam line and used for warming up the engine after it has been shut down for any length of time.

76. Cylinder Ends. — The cylinder ends are sometimes jacketed, in which case the cylinder bottom and cylinder cover are made double, with the two walls stiffened by occasional ribs. If the ends are not to be jacketed they can be made with a single wall stiffened by deep ribs. The cover and bottom are occasionally made double merely for the sake of stiffness.

77. Sizes of Parts. — The thickness of the liner and of the barrel can be computed by means of the formula given below:

$$t = \frac{(P + 25)\,D}{6000} + \frac{40}{100 + D}. \tag{39}$$

t = thickness of liner.

P = maximum pressure in cylinder.

D = diameter of cylinder in inches.

The value of P can be taken as follows:

Triple and Quad. H.P. cylinder P = boiler pressure (gage).
Triple . . . . M.P. cylinder $P = 0.5$ boiler pressure (gage)
Triple . . . . L.P. cylinder $P = 0.375$ boiler pressure (gage).
Quadruple . . . . . . I.M.P. cylinder $P = 0.6$ boiler pressure (gage).
Quadruple . . . . . . 2 M.P. cylinder $P = 0.4$ boiler pressure (gage).
Quadruple . . . . L.P. cylinder $P = 0.25$ boiler pressure (gage).

It is customary to make the liners of all cylinders of the same thickness and as the liner for the H.P. cylinder usually figures out to be the largest it is sufficient to calculate that one alone and make the others of the same thickness. Below is given a table of liner thickness for various sized H.P. cylinders:

TABLE 8

Diameter	Boiler pressure			Diameter	Boiler pressure		
	175	190	210		175	190	210
16	$\frac{7}{8}$	$\frac{15}{16}$	1	30	$1\frac{5}{16}$	$1\frac{3}{8}$	$1\frac{1}{2}$
17	$\frac{15}{16}$	$\frac{15}{16}$	1	32	$1\frac{3}{8}$	$1\frac{7}{16}$	$1\frac{9}{16}$
18	$\frac{15}{16}$	1	$1\frac{1}{16}$	34	$1\frac{7}{16}$	$1\frac{1}{2}$	$1\frac{5}{8}$
19	1	1	$1\frac{1}{16}$	36	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$
20	1	$1\frac{1}{16}$	$1\frac{1}{8}$	38	$1\frac{9}{16}$	$1\frac{11}{16}$	$1\frac{13}{16}$
21	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	40	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$
22	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{3}{16}$	42	$1\frac{11}{16}$	$1\frac{13}{16}$	$1\frac{15}{16}$
23	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	44	$1\frac{3}{4}$	$1\frac{7}{8}$	2
24	$1\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{4}$	46	$1\frac{13}{16}$	$1\frac{15}{16}$	$2\frac{1}{16}$
25	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	48	$1\frac{7}{8}$	2	$2\frac{1}{8}$
26	$1\frac{3}{16}$	$1\frac{1}{4}$	$1\frac{5}{16}$	50	$1\frac{15}{16}$	$2\frac{1}{16}$	$2\frac{1}{4}$
28	$1\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{3}{8}$	52	2	$2\frac{1}{8}$	$2\frac{5}{16}$

When a liner is used the barrel can be made $\frac{1}{8}$ inch thinner than the liner since the liner must have some extra thickness to allow for reboring. When no liner is used it is safer to make the barrel thickness $\frac{1}{8}$ inch greater than the liner thickness given in the table, as a cracked cylinder barrel is difficult to replace, while a liner can easily be renewed.

The thickness of the different walls are about as follows:

Thickness of metal in liner $= t$ (see table above).
Thickness of metal in barrel $= t - \frac{1}{8}''$ (with liner)
$= t + \frac{1}{8}''$ (without liner).

Thickness of metal in cylinder bottom = t (single)
= $0.9\,t$ (double).
Thickness of metal in cylinder cover = t (single)
= $0.85\,t$ (double).
Thickness of metal in steam passage = $0.85\,t$.
Thickness of metal in valve liners = t.
Thickness of metal in cylinder feet = t.
Thickness of metal in cylinder-feet flanges = $1.5\,t$ to $1.75\,t$.
Thickness of metal in cylinder-cover flange = $1.3\,t$ to $1.4\,t$.
Thickness of ribs in single cover or bottom = $t - \frac{1}{16}''$.
Depth of ribs in single cover or bottom = $5\,t$ (at least).
Distance between double walls of cover or bottom = $5\,t$ (at least).
Width of cylinder cover joint = $2.75\,t$ to $3.25\,t$.
Diameter of bolts in cylinder feet = $1.4\,t$ to $1.6\,t$.
Width of jacket space = $\frac{3}{4}$ to 1 inch.

Spacing of ribs in cover and bottom = $\dfrac{100\,t}{\sqrt{P}}$.

78. Attachment of Liner. — The liner can be held in place in several ways. One very common method is to have a flange at the lower end of the liner which is bolted to the cylinder bottom, see Fig. 39. This construction makes the ports longer than necessary and can be avoided by cutting off the flange in the way of the port and bolting the liner through the barrel. Since this puts the bolts in shear it is not good practice. The flange can be dispensed with entirely and the liner extended to the cylinder cover; thus the liner is held between the cover and the bottom of the cylinder without the use of any bolts. (See Plates 2 and 3.) In cases where the liner does not extend to the cylinder cover a special packing is

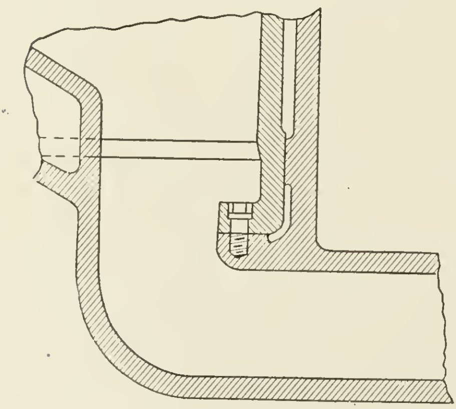


FIG. 39.

provided to prevent leakage of steam at the joint between the barrel and the liner. The liner is usually counterbored at the bottom and top so that the piston rings will over-travel. The amount of over-travel at the top should be about $\frac{1}{4}$ inch and at the bottom it can be $\frac{1}{8}$ inch.

79. Piston Clearances. — The shape of the cylinder bottom and the under side of the cover must conform to the shape of the piston. The length of the cylinder must be such that the linear clearances at the top and bottom will be about as follows:

Diameter of L.P. cylinder	Top clearance	Bottom clearance
40 inches and below........	$\frac{3}{8}$ inch	$\frac{1}{2}$ inch
60 inches..................	$\frac{1}{2}$ inch	$\frac{5}{8}$ inch
80 inches..................	$\frac{5}{8}$ inch	$\frac{3}{4}$ inch
100 inches and above........	$\frac{3}{4}$ inch	$\frac{7}{8}$ inch

80. Ports and Passages.— The ports leading from the cylinder to the valve chest should be short and direct in order that the clearance volume may be as small as possible. In earlier designs the ports were made long and curved to bring the valve-chest cover joints to the same level as the cylinder-cover joints. In more recent design the ports are made inclined or horizontal and the valve chest extends above and below the cylinder. The distance between the end of the piston-valve liner and the valve-chest cover must be sufficient to allow a clear steam passage with an area at least equal to that of the receiver pipe communicating with that end of the valve.

The height of the steam passage between the cylinder and the valve chest will depend upon the valve arrangement and the steam speeds used. It is commonly made not less than 3 inches and may vary from that up to 6 inches. This height is usually increased about 1 inch in the M.P. cylinder and about 2 inches in the L.P. cylinder. The cylinder cover joint is usually at the same height in all cylinders and is so placed that in the L.P. cylinder, or whichever cylinder has the steam passage of maximum height, there shall be from 3 to 4 inches of metal above the passage to allow the cylinder cover studs to be put in without

breaking through into the steam passage. The cross-sectional area of the steam passage from the valve to the cylinder should be equal to the port area through the valve liner. The wall of the passage around the valve is generally circular in shape and eccentric with the valve liner. The width at the narrowest part is made not less than $1\frac{1}{2}$ inches and at other points should be sufficient to keep the steam speed constant.

81. Cylinder Openings. — Various openings into the cylinder and valve chest must be provided. In the ends of each cylinder provision must be made for relief valves, and in the lower end there must be an additional opening for a cylinder drain. This drain should be placed in the lowest part of the steam passage. The relief valve in the lower end must be so located that it will not foul the connecting rod at the top of its stroke. The indicator bosses must be so placed at the top and bottom of the cylinder that the openings are not covered by the piston at the top and bottom dead points, and they must be far enough from the steam ports to prevent the pressure from being affected by the velocity of the steam. If jackets are used bosses must be provided for steam pipes leading to and from them, and also for a drain. At some convenient point in the steam passage "peep holes" must be located to give a view of the openings in the valve liner, by means of which the setting of the valve can be determined. There must be an opening for a bottom drain in the valve chest and relief valves are placed either on the chest or on the receiver pipes. Whenever the bosses are on the barrels they must be carried out beyond the line of the lagging.

82. Cylinder Feet. — The distance from the horizontal center line of the cylinders to the cylinder feet is made the same in all cylinders. In the case of the L.P. cylinder the feet are usually covered by the cylinder bottom and consequently must be located sufficiently below the lower lagging flange to give access to the bolts and nuts making up the joint. A distance of 8 inches from the lower lagging flange to the top of the foot flange will be sufficient for this purpose. The feet on the other cylinders are placed at the same level as those on the L.P. cylinder.

83. Boring-bar Opening. — The opening in the cylinder bottom must be large enough to permit the use of a good sized boring bar. In the H.P. cylinder the opening is made as large as the construction will permit. The openings in the M.P. and L.P. cylinders are usually made equal to about one-fourth the diameter of the L.P. cylinder. The covers for these holes carry the piston-rod packing and should be made deep enough for this purpose.

84. Cylinder-cover Studs. — There are two conditions which determine the size and number of cylinder cover studs: they must have enough strength to carry the load on the cover, and they must be spaced closely enough to make the joint steam tight. It is usual to make the studs of the same size on all the covers and to determine the size from the load on the H.P. cylinder; the spacing on the other cylinders can be increased on account of the decreased steam pressure. The maximum load upon the H.P. cylinder cover can be found by multiplying the boiler pressure by the maximum area of cover exposed to steam. The bore of the cylinder at the top will vary from $Hd + \frac{1}{4}$ inch to $Hd + \frac{1}{2}$ inch according to the size of cylinder. Using the larger value the load upon the cover will be

$$\frac{\pi}{4}\left(Hd + \frac{1}{2}\text{ inch}\right)^2 \times \text{gage boiler pressure.}$$

About the smallest stud used in cylinder covers is 1 inch in diameter and for any given cylinder the stud diameter will generally be slightly greater than the thickness of the liner. The diameter d can be arbitrarily chosen and from Table 5 the working load of that size stud can be found. The total load on the cylinder cover divided by the working load of one stud will give n the number of studs. The pitch circle for the studs will lie in the middle of the joint and if it has a width of 3 d the pitch circle will have a diameter

$$F = Hd + \tfrac{1}{2}\text{ inch} + 3\,d. \tag{40}$$

$$Hd = \text{diameter of H.P. cylinder.}$$

The pitch of the studs will be $\frac{\pi F}{n}\cdot$

The pitch of studs on the H.P. cylinder should be from $2.75\,d$ to $3.25\,d$. If they are closer than $2.75\,d$ it will be difficult to get a spanner on, and if the spacing exceeds $3.25\,d$ the joint is apt to leak. If, with the value of d chosen, the spacing is too small the value of d must be increased; if the spacing is too large, the value of d must be decreased. On the M.P. and L.P. cylinder covers use the same sized stud, spaced $4\,d$ to $4.5\,d$ on the M.P., and $5\,d$ to $5.5\,d$ on the L.P. cover. The diameter of the outside of the cylinder cover will be

$$G = Hd + \tfrac{1}{2}\text{ inch} + 6\,d, \tag{41}$$

if the width of the joint is taken as $3\,d$.

85. Valve-chest Cover and Studs. — The valve-chest covers are usually held by 1-inch studs spaced from 4 to 5 inches apart and the width of the joint can be made 3 inches. The diameter of the opening through the upper and lower end of the valve chest must be large enough to pass the valve liner. If the diameter of the valve is v and the thickness of the liner is t the opening for the valve-chest cover will be about

$$v + 2\,t + \tfrac{1}{2}\text{ inch},$$

and the diameter of the valve-chest covers will be

$$v + 2\,t + 6.5\text{ inches}.$$

When twin valves are used the distance between centers should not be less than $1.6\,v + 1$ inch.

MAIN BEARINGS

86. Character of Loads upon Bearings. — The diameter of the crank shaft in the main bearings is given by Formula (20). The length of any bearing is made such that the bearing pressure shall not exceed that given in Table 6. The loads to which the main bearings are subjected increase as we go towards the propeller, and differ in character. In some cases the bearing pressure is well distributed over the entire circumference of the bearing; in other cases it tends to act upon a small portion only. Since the important thing about a bearing is that it shall be sufficiently large to keep cool, it can be seen that a larger unit bearing

pressure can be allowed where the load is well distributed than where it is concentrated upon one part only.

When a piece of shafting, such as a line shaft, for instance, is transmitting power there is no load upon the supports except that due to the weight of the shafting. If, however, a crank is introduced into the shaft, and a bearing is placed on each side of the crank, these bearings will be subjected to loads in addition to those from weight alone. The force which one web delivers through the crank pin to the other web will be felt upon the bearing adjacent to the latter web, while the reaction which the first web experiences will be felt upon the bearing nearest that web. These loads upon the bearings must be equal in amount and opposite in direction, in order that the shaft which was in equilibrium before the crank was introduced may still be in equilibrium.

The turning force acting at the crank pin will increase as we go towards the propeller, and, in consequence, this component of the bearing pressure will become more predominant. The components of the bearing pressure are shown in Fig. 40. OA is the position of the crank after it has turned through 150°. AB is the force acting through the connecting rod, and is composed of the weight and inertia of the reciprocating masses and the unbalanced steam pressure upon the piston. BC is the cross-throw of the connecting rod. CD is the resultant of the weight of the shaft and the weight and inertia of the connecting rod. DE is the centrifugal force arising from the rotation of the unbalanced parts of the crank webs and pin. EF is the force acting upon the aft web, due to the turning force from the forward cylinders. EG is the reaction upon the forward web, due

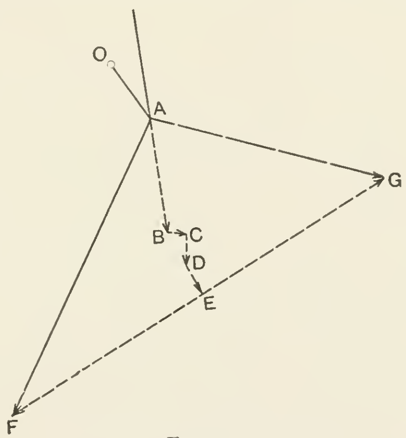


FIG. 40.

to the above force. AF is the resultant force acting upon the aft bearing, and AG is the resultant for the forward bearing.

If the turning force EF and the reaction EG were left out, as is the case in the bearings of the first cylinder, the two bearings would be subjected to the same load AE. The effect of the turning force, however, is to cause the resultant loads to differ quite widely in amount and in direction. Since the turning force varies its direction through 360° during a revolution, its effect upon the resultant as it becomes more predominant in the bearings of the crank shaft nearer the propeller, is to cause the load to be distributed more uniformly around the bearing. In the case of the forward bearing of each pair of bearings, since the reaction of the turning force is, in general, opposed to the direction of the other forces, the resultant tends to act upon the side of the bearing. This is shown very clearly in Fig. 41. The full lines give the direction and intensity of the loads upon the aft bearings and the broken lines are for the forward bearings.

87. Loads upon Main Bearings. — The mean load upon the bearings can be found approximately by means of the formula

$$L = \frac{21{,}000}{\text{P.S.}}[\text{H.P.}_T + a\text{H.P.}_C]. \tag{42}$$

P.S. = piston speed in feet per minute.

H.P._T = indicated horse-power developed forward of cylinder whose bearings are in question.

H.P._C = indicated horse-power developed in cylinder over the bearings.

a = a factor whose value can be taken from the curves given in Fig. 42.

The factor a allows for two components of the bearing load, one due to the power being developed in the cylinder over the bearing, and the other due to the centrifugal force of the rotating parts. If we had to deal with the first component only, a would be constant for all sized engines, but since the centrifugal force of the rotating parts increases at a greater rate than the horsepower of the engine, the factor a will increase with the size of the engine.

Diagrams Showing Intensity, Direction and Distribution of Loads on the Main Bearings of the Engines of U.S.S. Monterey

Full Lines are for Aft Bearings of Cylinder.

Broken Lines are for Forward Bearings of Cylinder.

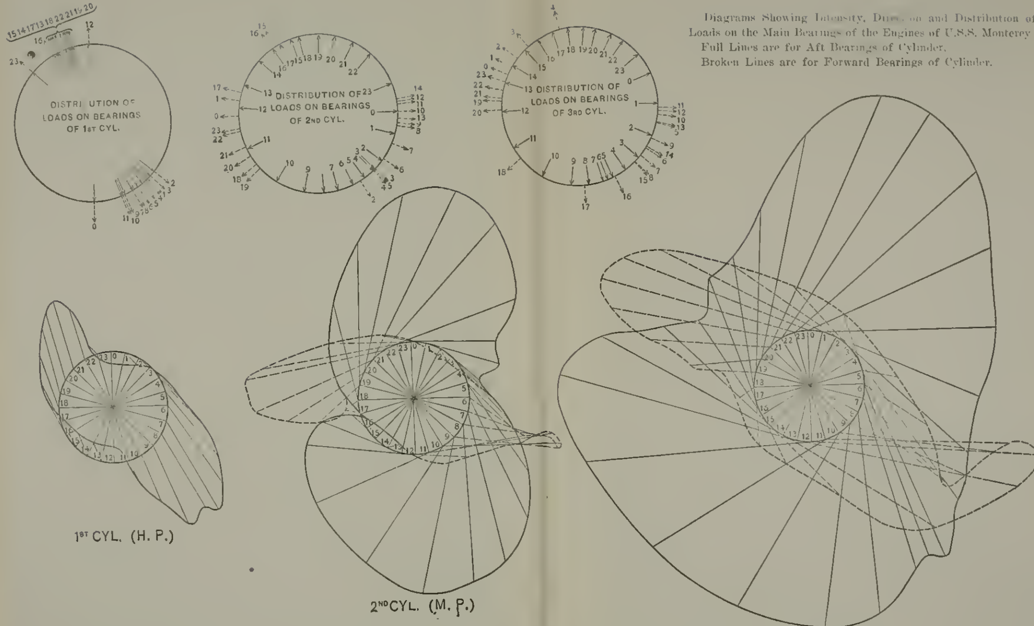


FIG. 41

Mean Load on Bearings $= \frac{21{,}000}{P.S.}(H.P._T + a\ H.P._c)$

$H.P._T$ = Horse Power Transmitted from Forward Cylinders

$H.P._c$ = " " Developed in Cylinder over Bearing

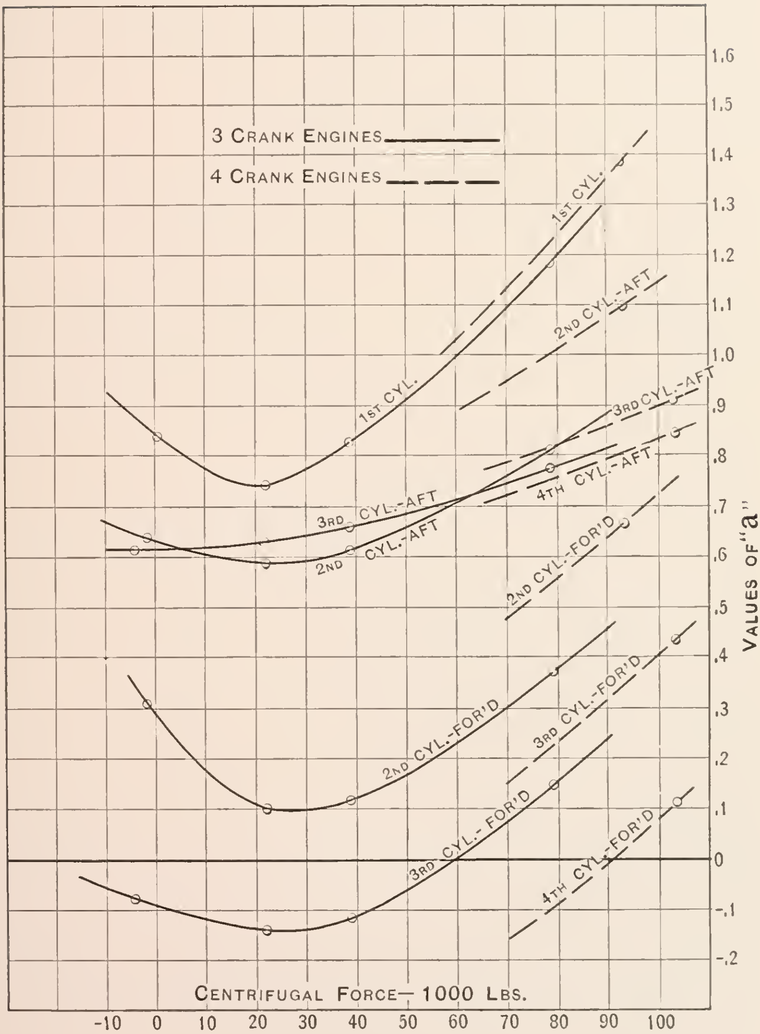


Fig. 42.

88. Centrifugal Force of Crank. — The centrifugal force exerted by the rotating parts of one crank will be given approximately by the following formula:

$$F = C\,(\text{I.H.P.})\,n^2S. \tag{43}$$

I.H.P. = *total* I.H.P. of engine.
n = revolutions per second.
S = stroke in feet.
C = 0.5, solid forged shaft
= 0.83, built-up shaft, Triple engine
= 1.1, built-up shaft, Quadruple engine.

89. Combined Bearings. — The two bearings between adjacent cylinders are sometimes combined into one bearing. In this case the load on the common bearing will be a certain fraction of the sum of the loads which would act upon the separate bearings. This fraction of the total load is given in the following table:

TABLE 9

Bearing between:	Type of engine	Angle between cranks	Factor
1st and 2d cylinders..	Triple, high leading	120°	0.6
	Quadruple	180°	0.45
	Triple, low leading	240°	0.85
2d and 3d cylinders..	Quadruple	90°	0.47
	Triple, high leading	120°	0.625
	Triple, low leading	240°	0.9
3d and 4th cylinders..	Quadruple	180°	0.875

When two cranks make an angle of 180° with one another, with a common bearing between them, the steam loads act in opposite directions, and balance, if equal powers are developed in the two cylinders. On the other hand the loads from the turning forces transmitted through the crank pins are added to one another since the common bearing serves as the aft bearing of one cylinder and the forward bearing of the other. Since the cranks make an angle of 180°, the two forces, which in the bearings of a single crank are opposed, are in this case added to one another.

The length of the main bearings can be determined from the

load per square inch that each bearing can carry, and these loads are given in Table 6.

90. Crank-pin Load. — The load acting upon the crank pins will be of about the same character as those upon the bearings of the first cylinder. The magnitude of the mean load will be given by the formula

$$L_1 = \frac{21{,}000}{\text{P.S.}} 1.6 \times \text{I.H.P.} \tag{44}$$

P.S. = piston speed in feet per minute.
I.H.P. = indicated horse-power developed in cylinder over crank pin.

The allowable pressure per square inch upon the crank pins can be from 200 to 250 pounds in merchant engines, and from 300 to 350 pounds in naval engines.

ENGINE FRAMING

91. Cylinder Supports. — The cylinder supports are of four kinds: the inverted Y hollow casting, see Plate 1; the straight box casting, see Plate 2; the straight column of I-section; and the steel column, see Plate 1. The supports for a given engine may be all of one kind or a combination of the different types. The framing as a whole can be divided into five classes depending upon the combination of these different types. (1) Plate 3 shows an engine with the condenser forming a part of the back framing, the guides carried by short box castings resting on the condenser top, and the front of the engine supported by long box castings. (2) Plate 2 shows an engine with the cylinders supported by straight box castings at front and back, the back supports carrying the guides. (3) Plate 1 shows a more open type of framing with inverted Y castings at the back and steel columns at the front. (4) Engines with large, heavy cylinders are sometimes supported by four castings, usually of the open I-section. The four-slipper type of crosshead is used with this framing. (5) This class includes most naval and yacht engines and the framing is as shown by Plate 4. These columns and tie rods are of wrought steel and give a strong, rigid, but expen-

sive, construction. The guides are carried by castings bolted to the fore and aft tie rods, or to the bottom of the cylinders and to tie rods. The box castings of the first three classes are usually made of cast iron, and cast steel is used for the fourth class. Where lightness is necessary, as in naval engines, the castings of the first three classes may be of steel.

The thickness of metal for the castings is not determined by the question of strength alone; castings must be rigid and thick enough to cast well. When made of cast iron the thickness is from 1 to $1\frac{1}{4}$ inches; when made of cast steel, from $\frac{3}{4}$ to 1 inch. If the thickness is determined from the load that comes through the connecting rod the working stress should be about 600 pounds for cast iron and about 1000 for cast steel. The size of the wrought-steel columns can be determined from the Column Formula (12), or taken from the curves of Fig. 22. The length of the columns should be taken from the assembly drawing but in the case of merchant engines will be from $3 \times S$ to $3.25 \times S$. The load used in figuring them should be the entire load W not $\frac{W}{2}$, since the rolling and pitching of the ship puts additional stresses on the columns due to the inertia of the heavy cylinders.

92. Column Flanges. — The flanges at the foot of the cast columns should have a thickness of about 2.25 times the thickness of metal, and the flanges at the top should be of the same thickness as the flanges of the cylinder feet. The wrought-steel columns are attached to cylinders and bed either by lugs, as in Plate 1, or by a circular flange. In the latter case four bolts are used and in the former two. The thickness of lug or flange should be slightly greater than the diameter of bolt going through it, and the bolts should have enough strength to carry the load W. The width of the lugs should be sufficient to provide bearing surface for the bolt heads.

93. Cylinder-column Bolts. — The diameter of the bolts in the cylinder feet should be from 1.4 to 1.6 the thickness of the cylinder liner and there should be enough bolt strength to carry the load W. The dimensions of the box casting at the top should be determined by the breadth of the guide which the column is

to carry and there should be enough flange space for the bolts required. The bolts in the top flange are spaced about 3 diameters apart and the width of the joint is about 3 diameters. The bottom flange of the cylinder supports carries about 50 per cent more bolts than the top flange and the spacing is about 4 diameters.

94. Engine Beds. — The beds are made of cast iron and cast steel and have about the same thickness as the cylinder supports. The bed consists of two longitudinal girders with cross girders

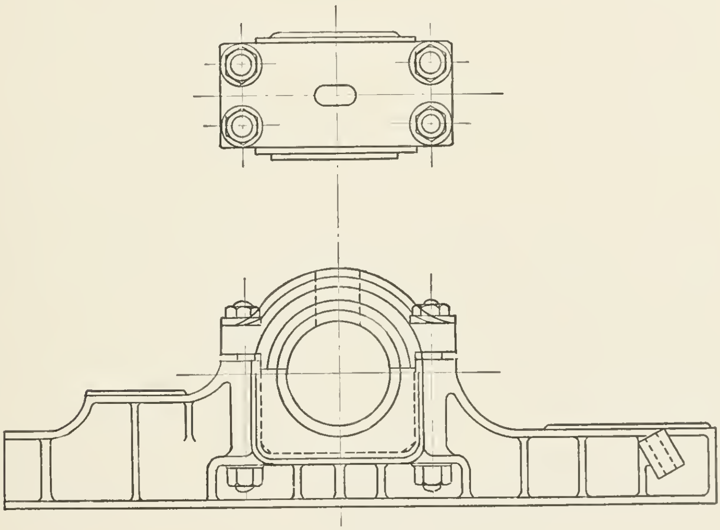


FIG. 43.

under each main bearing. The shape of the girder depends upon the material used; if made of cast iron they are of the hollow box form shown in Plates 1, 2, and 3; if made of cast steel the open I-section is used, see Fig. 43. The bed is usually carried down so that the bottom of the holding-down flange is below the lowest point in the clearance diagram of the connecting rod. This will bring the holding-down flange about 0.85 stroke below the center line of the crank shaft. If desirable it can be cut away at the sides as shown in Plate 2. The distance from the top of the side girders to the lowest point of the bed is about 0.55

stroke for cast-iron beds, and about 0.5 stroke for cast-steel beds. The breadth of the cross girders is always less than the length of the bearings which they support. The breadth must not be made so small that a good casting cannot be obtained, and for this reason it is not always possible to make the bearings for the first cylinder as short as the load will permit. The breadth of the cross girder should be at least 3 inches less than the length of the bearing, which it supports. The cross girders should be placed as close to the crank webs as possible in order that the bending moment on the shaft may be a minimum. In the engine shown on Plate 3 it will be noticed that the valves have been so placed that the Marshall valve gear eccentric has to come between the bearing and the crank web.

95. Main-bearing Bolts. — The main-bearing caps are held by two or more bolts, and the size of these bolts should be determined by the maximum load to which the cap is subjected. The mean loads were used in obtaining the sizes of the main bearings, but the maximum loads must be used in finding the thickness of the bearing caps and the sizes of the bolts. The maximum load upon any bearing can be found by multiplying the mean load, as given by Formula (42), by the following factors:

	Triple	Quadruple
1st cylinder, forward bearing, factor =	2	2
aft bearing, factor =	2	2
2d cylinder, forward bearing, factor =	1.75	1.75
aft bearing, factor =	1.75	1.75
3rd cylinder, forward bearing, factor =	1.67	1.75
aft bearing, factor =	1.4	1.75
4th cylinder, forward bearing, factor =	....	1.67
aft bearing, factor =	....	1.4

It is best to use one size of bolt for all the bearings.

Where there is a common bearing between two cranks the cap is often so broad that four bolts are required to hold it firmly in place. The maximum load to which any pair of bolts will be subjected will be twice the mean load when two bolts are used at the center of the bearing, and 1.67 times the mean load when four bolts are used, one pair at each end of the bearing.

When the main bearing is of the type shown in Plates 1 and 3, the bolts can be placed close to the shaft, clearing it by $\frac{3}{4}$ to 1 inch. When the type shown in Plate 2 is used the distance between the bolts will be from 1.5 the diameter of the shaft to 1.75 the diameter.

96. Main-bearing Caps. — The cap should be figured as a beam subjected to a bending moment of $\frac{Wl}{6}$, where W is the maximum load upon the bearing cap and l is the distance between the bolts. The breadth of the cap will be 2 or 3 inches less than the length of the bearing, or equal to the breadth of the cross girder supporting the bearing. In calculating the moment of inertia of the section allowance should be made for the hand-hole which is usually placed in the cap. The hand-hole is about $2\frac{1}{2}$ inches broad by 5 inches long. The load upon the cap is intermittent and a working stress factor of about 10 should be used for the sake of stiffness.

The caps may be made of wrought steel or cast steel. In some cases cast iron is also used. If wrought steel is used the top brass is made separate and the distance from the center line of the shaft to the back of the brass will be about 0.67 of the diameter of the shaft. Where cast steel or cast iron is used the thickness of the cap must be increased by $\frac{3}{4}$ or 1 inch to allow for white metal.

CYLINDER ARRANGEMENTS

97. Sequence of Cylinders. — The arrangement of the cylinders of an engine having three or more cranks will depend largely upon the question of balance. Some compound engines are made with one high-pressure cylinder and two low-pressure cylinders, in order that the crank may be placed at 120° to give a uniform turning moment. Some of the inclined engines used upon paddle wheel steamers have this arrangement.

The cylinders and valves of a three-cylinder Triple are usually arranged as shown on Plate 1. In some cases the low-pressure cylinder is placed in the center to give a better balance. Plate 2

shows a compact design with the valves placed at one side of the cylinders and run by Marshall valve gears.

The four-cylinder Triple is a common type for naval engines. The two low-pressure cylinders usually develop less power than the others and their reciprocating parts are made lighter. These lighter parts should be placed at the ends of the engine, so that we have the following arrangement for balanced four-cylinder Triples, — forward L.P., H.P., M.P., and aft L.P.

Quadruple engines are often arranged with the cylinders in the sequence of their sizes; H.P., 1 M.P., 2 M.P., L.P. An arrangement which gives better balance is to have the H.P. cylinder at one end and the 1 M.P. at the other with the 2 M.P. and L.P. in the middle. Six-cylinder Quadruples are occasionally used on high-powered vessels and the usual practice is to divide the H.P. cylinder into two cylinders, and the L.P. into two cylinders. These are placed tandem in the middle of the engine and the 1 M.P. and 2 M.P. cylinders are placed on the ends.

98. Space Occupied by Engines. — It is often convenient to know approximately how much space an engine will take up. It will be found that with the usual arrangement of cylinders and valves on the center line of the engine, the length over the cylinders will be about $1.9\,\Sigma D$ for merchant engines and about $1.8\,\Sigma D$ for naval engines. Where the valves are not on the center line the length will be about $1.65\,\Sigma D$. ΣD is the sum of the diameters of the cylinders. The length of the engine bed will be about 0.85 of the length over the cylinders. The height of the engine from the lowest point of the bed to the top of the cylinder covers will be about $5 \times S$ for naval engines where $\frac{l}{r} = 4$, and from $5.5 \times S$ to $6 \times S$ for merchant engines where $\frac{l}{r}$ has a value varying from 4.2 to 4.5. S = stroke of the engine. The breadth of the engine bed will be equal to about twice the diameter of the L.P. cylinder in merchant engines and about 1.75 that diameter in naval engines.

VALVE DIAGRAM

99. Eccentricity. — The quantities usually known or assumed before starting the valve diagram are the mean cut-off, eccentricity, leads, and steam speeds. The mean cut-off is determined by the desired distribution of power and the other quantities are assumed. The eccentricities used upon marine engines vary from 3 to $5\frac{1}{2}$ inches, depending upon the size of engine and the steam speeds employed, and will have about the following values for different sized engines:

I.H.P. of engine	500 to 1000	1000 to 2000	2000 to 5000	5000 to 10,000
Eccent. of H.P. and M.P. cylinders	3 to $3\frac{1}{2}$	$3\frac{1}{2}$ to 4	4 to $4\frac{1}{2}$	$4\frac{1}{2}$ to 5
Eccent. of L.P. cylinder	3 to $3\frac{3}{4}$	$3\frac{1}{2}$ to $4\frac{1}{4}$	4 to 5	$4\frac{1}{2}$ to $5\frac{1}{2}$

100. Steam Speeds. — The steam speeds in feet per minute should be about as follows:

	Merchant	Naval	
Main steam pipe..........	6500	7000	
Throttle valve............	6000	6500	
	H.P.	M.P.	L.P.
Entering steam............	6000	7500	10500
Exhaust steam............	5000	6500	7500
Exhaust to condenser......	6500	....	....

In some naval engines the steam speeds at maximum power will be slightly higher than these.

101. Width of Ports. — The maximum port opening will be determined from the valve diagram, and the width of port will be determined by the ratio of the speed of the entering steam to the speed of the exhaust steam.

Width of port =

$$\text{mean of maximum port openings} \times \frac{\text{speed of entering steam}}{\text{speed of exhaust steam}}$$

The width of port should be from 0.6 of the eccentricity to 0.75 of the same, although if the cut-off is quite long the ratio

may run as high as 0.85. If the width does not come within these limits the assumed values of the steam leads should be changed.

102. Steam Lead. — The valves of marine engines are designed with greater steam leads than those of land engines. The ratio of steam lead to eccentricity is ordinarily 0.13 for the top end of the H.P. cylinder valve, and about 0.16 for the top end of the L.P. cylinder valve. The ratio may be greater than these if the cut-off is short. The steam lead on the bottom end of the valve is usually greater than that on the top by an amount varying from $\frac{3}{16}$ to $\frac{5}{16}$ inch, for valves taking steam on the ends, and greater by an amount varying from $\frac{1}{16}$ to $\frac{1}{8}$ inch for valves taking steam in the middle. The difference is made less in the latter case because the wear of the joints in the valve gear increases this difference, while in valves taking steam on the ends the effect of wear is to decrease the original difference. The valve of the H.P. cylinder should always take steam in the middle, while in the L.P. cylinder the valves should take steam on the ends. The intermediate valves may be made either way.

103. Size of Piston Slide Valve. — The diameter of the piston slide valves can be found by the following formula:

$$d = \frac{D^2 \times \text{P.S.}}{4\,cXw}. \qquad (45)$$

d should not be less than $4\,c\,w$.
d = diameter of valve in inches.
D = diameter of cylinder in inches.
P.S. = piston speed in feet per minute.
c = portion of entire circumference of liner available for clear port opening, and is about 0.75 in merchant engines and about 0.85 in naval engines.
X = speed of exhaust steam in feet per minute.
w = width of port in inches.

If d is less than $4\,c\,w$ the area bounded by the inner circumference of the liner will be less than the area through the ports.

104. Size of Flat Slide Valve. — The breadth of the flat slide valve can be found by means of formula (45) if we multiply both sides by π and let $c = 1$.

$$b = \frac{\pi D^2 \times \text{P.S.}}{4 Xw}. \qquad (46)$$

b = clear breadth of port of flat slide valve.

It will be necessary to put in 1-inch ribs about every 16 inches, so the total breadth will have to be increased to allow for these

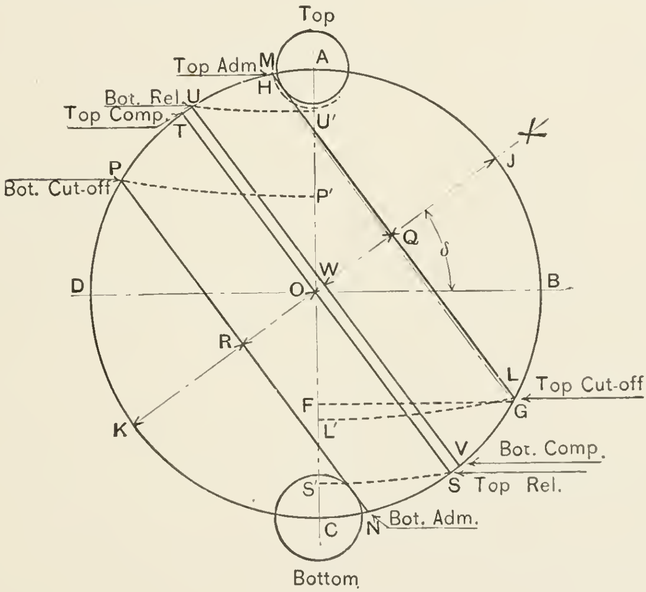


FIG. 44.

ribs. If the breadth is more than 0.95 D it will be necessary to use a double-ported valve, and if this makes the valve long and narrow the eccentricity, and consequently w, can be decreased.

105. Valve Diagram. — The valve diagram from which the maximum port opening, angle of advance, steam laps, and exhaust laps are determined can be drawn as shown in Fig. 44. The circle $ABCD$ has a radius equal to the eccentricity. Place the point F so that $\dfrac{AF}{AC}$ = mean cut-off. Erect the perpendicular FG. At A describe an arc whose radius is equal to the

mean of the steam leads of the two ends. Draw GH tangent to this arc and draw a line JK through O perpendicular to GH. The angle BOJ will be the angle of advance. At A and C draw the lead circles as shown in full lines, the radius of each being equal to the steam lead at that end. Draw the lines LM and NP tangent to the lead circles and perpendicular to JK. Draw lines OL, OM, ON, and OP. These lines represent the crank position for admission and cut-off at top and bottom. OQ and OR will be the steam laps and QJ and RK the maximum port openings. The percentage of the stroke traversed by the piston at the time of cut-off can be determined by swinging from L and P with a radius equal to $OA \times \frac{l}{r}$.

The release and compression of the steam on the top side of the piston will occur at about the right time relative to the other events if the exhaust lap is made zero, thus causing the events to occur at mid-position of the valve, OS and OT. The release of the steam on the under side of the piston should occur 2 or 3 per cent of the stroke earlier than the release on the top side; i.e.,

$$\frac{AU'}{AC} = \frac{CS'}{AC} + 0.02.$$

UV is perpendicular to JK and the distance OW is the exhaust lap for the bottom of the valve.

The relation between the angle of advance, eccentricity, mean cut-off, and mean lead is given by the following equation:

$$\cos \delta = \frac{1}{2e}\left[\sqrt{4e^2B - a^2} - a\sqrt{\frac{1-B}{B}}\right]. \tag{47}$$

e = eccentricity.
B = mean cut-off (decimal).
a = mean lead.

The results obtained graphically should be checked by the above formula. It should also be true that steam lap + lead = $e \sin \delta$.

The results should be collected into a table of the following form:

TABLE 10

DIAMETER OF CYLINDERS AND STROKE

Scale of valve circle.

Ratio of connecting rod to crank $\left(\frac{l}{r}\right)$.

	H.P.		M.P.		L.P.	
Eccentricity......................	....		....		....	
Size and kind of valve.............	....		....		....	
Side of valve on which steam is taken	....		....		....	
	Top	Bottom	Top	Bottom	Top	Bottom
Width and length of port..........	...	....	...	....	...	....
Steam lap.........................	...	....	...	....	...	....
Exhaust lap.......................	...	....	...	....	...	....
Angular advance...................	...	....	....	....	...	....
Steam lead, linear................	...	....	...	....	...	....
Cut-off, in decimal of stroke......	...	....	...	....	....	....
Exhaust lead, in decimal of stroke.	...	....	...	....	...	....
Compression, in decimal of stroke..	...	....	...	....	...	....
Maximum port opening...........	...	....	...	....	...	....
Velocity of steam, feet per min....	...	....	...	....	...	....
Velocity of exhaust, feet per min...	...	....	...	....	...	....

When the valves take steam at the middle the valve diagram is drawn the same as for valves taking steam at the ends, but the eccentrics are set $270° +$ *angle of advance* ahead and back of the crank instead of $90° +$ *angle of advance.*

VALVES AND VALVE GEAR

106. Piston Valves. — Piston valves are made solid, as shown in Plate 2, or hollow, as shown in Plate 3. They are made hollow when it is desired to have one pipe supply steam to both ends of the valve, and the area through the middle of the valve should equal the area through the pipe. The length of the valve depends upon the location of the valve liners, which are placed a sufficient distance from the top and bottom of the valve chest to allow the steam to enter and get away from the ends without the area for steam passage being restricted. The liners should be placed as near the ends as possible, and the passage-ways to the cylinder made as direct as possible, in order to reduce the

clearance space. The length of the liners should be such that the piston-valve rings will not spring out at the extremities of the stroke. The liners should be counterbored at the ends sufficiently to allow the rings to over-travel $\frac{1}{4}$ inch or more. The length of the piston-valve liner = width of port + steam lap + exhaust lap + the travel of the valve.

107. Load upon Valve Stems. — The valve stems, eccentric rods, and links must be designed to take care of the frictional load, the inertia, and the weight of the valves. In addition, if a single valve is used with a guide, such as shown in Plate 2, the stem below the valve must be designed for the bending that may come upon it from the pull of the drag rods in reversing when the valve is at its lowest point in the stroke.

In the case of the flat valve, the frictional load can be calculated from the area of the surfaces in contact and the unbalanced pressure upon the back of the valve, no allowance being made for any balancing device. The piston valve is not subjected to any load due to unbalanced pressure, but the friction of the rings and the stuffing-box must be allowed for. It is usual to assume that this load is some multiple of the weight of the valve, valve stem, crosshead, and block. If this load is taken as three times the weight of the above parts, a reasonable allowance will be made.

The inertia of the valves is calculated upon the assumption of harmonic motion, and the maximum inertia, at the beginning and end of the stroke, is used. The formula which gives this inertia is

$$F = 0.000{,}028{,}37\ WRN^2, \tag{48}$$

where W = weight of valve or valves, valve stem, crosshead, and block, usually for the low-pressure gear;

R = eccentricity in inches;

and N = revolutions per minute.

In the case of piston valves, the load for which the valve stem should be figured will be

$$L = W\,(4 + 0.000{,}028{,}37\ RN^2), \tag{49}$$

if no balance piston is used; and

$$L = W\,(3 + 0.000{,}028{,}37\;RN^2), \tag{50}$$

if a balance piston is used.

If balance pistons are used which balance more than the weight the load can be still further decreased.

In the case of flat valves the load upon the valve stems will be:

$$L = pAf + W\,(1 + 0.000{,}028{,}37\;RN^2), \tag{51}$$

if no balance piston is used; and

$$L = pAf + 0.000{,}028{,}37\;WRN^2, \tag{52}$$

if a balance piston is used.

p = the unbalanced unit pressure upon valve;
A = area exposed to unbalanced pressure;
f = coefficient of friction, usually taken as 0.2;
W, R, and N are as above.

The portion of the valve stem between the valve and the link block in the case of single valves, and between the piston valve and the yoke in the case of twin valves, should be figured by means of the piston-rod formula; the portion of the stem within the valve should be figured for tension only, as the stem is shouldered down where it enters the valve, and the thrust is carried by this shoulder.

108. Valve-stem Bending. — The load L is carried by the links, and when the latter are at an angle with the horizontal (see Fig. 45) there will be a tendency for the block to slide along the links. As the valve stem is kept from moving by the valve stem guide, this tendency results in a bending moment upon the valve stem, and a reaction in the drag rods. The maximum angle that the links make with the horizontal is assumed to be

$$\sin^{-1} = \frac{2\,E \sin\,(90 - d)}{b}. \tag{53}$$

E = eccentricity of valve.
d = angle of advance.
b = distance between eccentric rod pins, usually $= 6\,E$.

The component which bends the valve stem will be

$$P = \frac{L\ 2\ E \sin\,(90^\circ - d)}{b}. \tag{54}$$

The bending moment upon the valve stem will be $M = Pl$, and l is the distance from the bottom of the valve stem guide to the

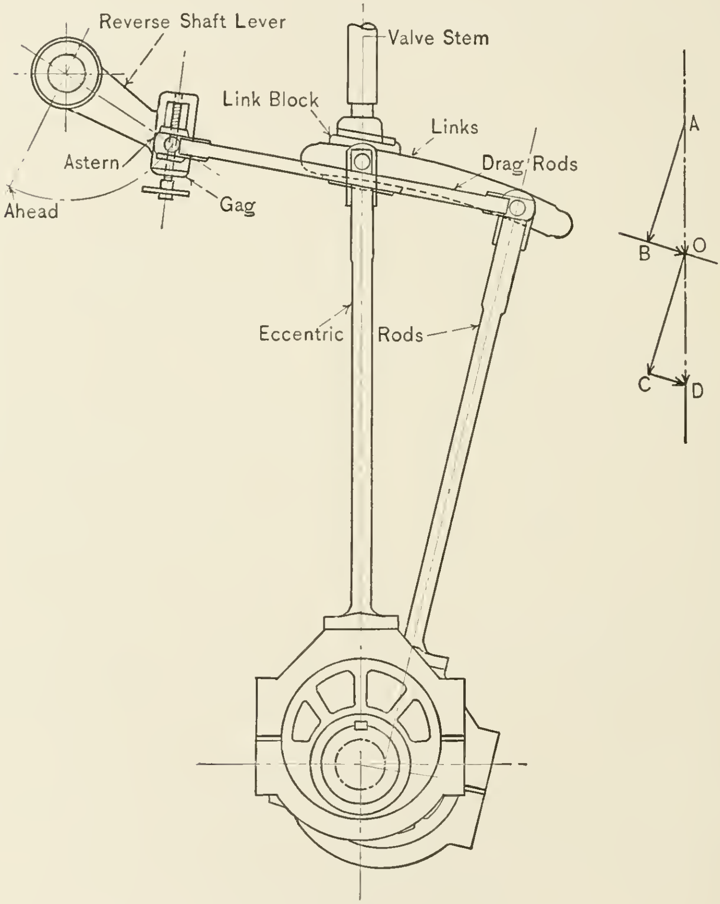


FIG. 45.

lowest point in the travel of the link block. For twin valves, with the stems yoked together, the bending moment would act upon the yoke, which always has plenty of strength.

109. Drag Rods. — If the eccentric rod were normal to the link at the time when the latter makes its maximum angle with the horizontal (see Fig. 45) the load P would be all that the drag rods would have to carry; but since the eccentric rod is in line with the valve stem at that time, a portion of the load normal to the links will come upon the drag rods in addition to the component along the links. In the diagram accompanying Fig. 45, AO is the load L acting through the valve stem. This load is resolved into BO along the links and AB normal to the links. Since the eccentric rod is in the position OD, the load $OC = AB$, normal to the links, is resolved into CD and OD. The drag rods have to take care of $BO + CD = 2\ P$, and the eccentric rod is subjected to the load L. Although the drag rods are not exactly parallel to the links in the position shown, the load upon the rods will be practically $2\ P$, so that each rod should be designed for the load P.

110. Yokes. — The yokes should be figured as beams supported at the ends and loaded at the middle with the load L. The surface of the valve-stem guide in the case of single valves, and of the valve-yoke guide in the case of twin valves, should be such as to keep the unit bearing pressure, due to the load P, between 70 and 100 pounds.

111. Eccentric Rods. — The diameter of the eccentric rods and drag rods at the middle should be figured by the connecting-rod formula, as they are columns hinged at the ends. The diameter of the drag rods at the ends can be made three-fourths of their diameter at the middle. The diameter of the eccentric rods at the top should be 0.9, and at the bottom 1.1 of the diameter at the middle. The length of the drag rods is usually from $15\ E$ to $18\ E$, and of the eccentric rods $20\ E$ to $30\ E$ for merchant engines and $15\ E$ to $20\ E$ for naval engines. The diameter of the bolts in the caps, etc., should be such as to carry the loads with the working stresses given in Table 5.

112. Link Bars. — The link is usually of the double-bar type, Fig. 46, and is figured as a beam supported at the ends and loaded at the middle with the load L; as mentioned before, the length of the beam, or distance between eccentric rod pins, a, is

usually 6 E. The breadth of the bars, b, is usually about one-third of the depth, c, and the depth is so chosen that the working stress factor will be 12, as the load is alternating. Making this assumption in regard to the relation of breadth and depth, the formula for the depth of each link bar becomes

$$h = \sqrt[3]{\frac{9\,La}{4f}}, \tag{55}$$

L = load upon valve gear.
a = distance between eccentric rod pins.
f = allowable stress with factor of 12.

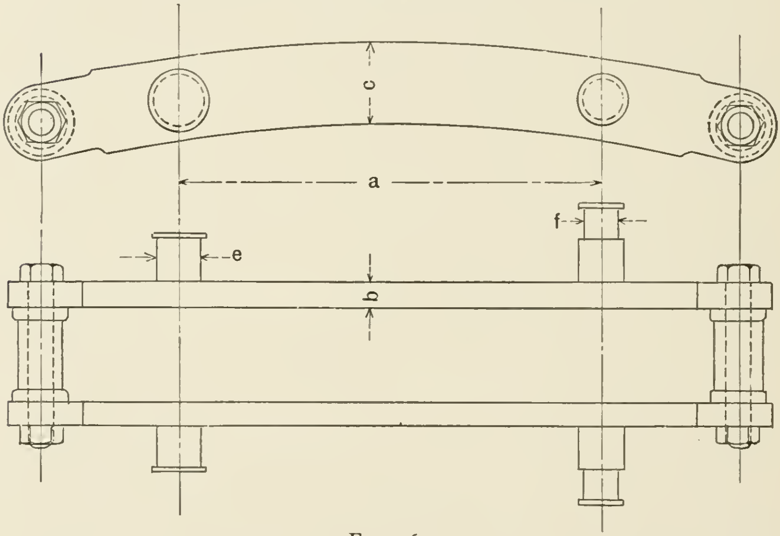


FIG. 46.

113. Link-block Pin. — The diameter d of the link-block pin, Fig. 47, is from 0.9 to 1.0 of the depth of the bar, c; the diameter of the eccentric-rod pins, e, Fig. 46, is about three-fourths the depth of the link bar, and the diameter of the drag-link pin, f, is about three-fourths the diameter of the eccentric-rod pin. The lengths of all these pins, as well as that of the block gibs, g, must be such as to keep the bearing pressures within the limits given in Table 6. The thickness of the metal, h, joining the link-block pin at the sides to the sliding gibs should be from 0.25 to 0.3 of the diameter of the link-block pin, d.

114. Eccentrics. — The diameter of the eccentric will be

$$D = 2\,(r + E + c). \tag{56}$$

E = the eccentricity.

r = radius of eccentric pad on crank shaft.

$c = \dfrac{r}{3}$, if the lower part of the eccentric is made of cast iron

$= \dfrac{r}{4}$, if it is made of cast steel.

The upper part of the eccentric, Fig. 48, is always made of cast iron, and is joined to the lower part by bolts or collar studs. The keyway should be cut on a line at right angles to the joint of the two parts, so that the eccentric can be readily taken off. If the eccentric is so situated that it can be moved along the shaft clear of the key, then the keyway can be on the side, as shown dotted in Fig. 48, and the set screw will be more conveniently located.

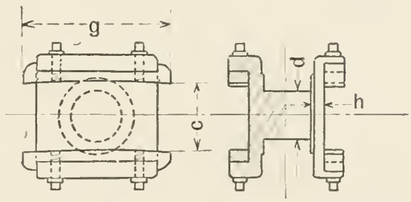


FIG. 47.

It is well to make the keyway considerably broader than the key in the shaft, and to fit liners on either side, so that slight changes can be made in the angular advance, if it is thought best. The breadth of the eccentric should be sufficient to keep the bearing pressure within the limits given in Table 6.

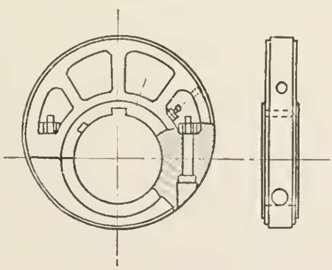


FIG. 48.

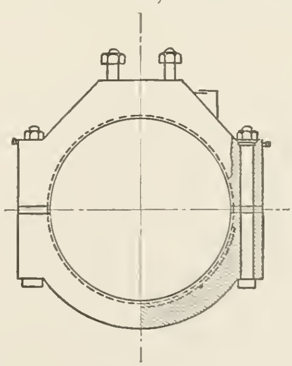


FIG. 49.

115. Eccentric Strap. — The eccentric strap, Fig. 49, should have lips fitting on each side of the bearing surface of the eccentric sheaves, to keep the strap in place. The strap bolts should be designed to carry the load L coming upon the valve gear, as should also the bolts uniting the eccentric rod to the strap. The straps are usually made of cast steel lined with white metal, and the section of the lower half, exclusive of the white metal, should be sufficient to carry the load L with a working stress factor of 8.

116. Reverse-shaft Levers. — The reverse-shaft levers, Fig. 45, should be figured for the load $2P$ upon the drag rods, and should be of such a length that the angle moved through is not more than 80°. The "gag" upon the lever should be so arranged that in the backing position the center line of the screw is vertical. This will cause the position of the link when backing to be practically the same, irrespective of the position of the end of the drag rod in the slot. In the ahead position of the lever the gag screw will be nearly horizontal, and the link can be pulled in an amount about equal to the travel of the nut on the thread.

117. Reverse Shaft. — The reverse shaft should be figured for torsion and bending, due to the thrust of the drag rods. As it is not always possible to get the bearings close to the various reverse shaft levers, the bending moment may be large in these shafts. The equivalent twisting moment should be found by Formula (15). The twisting moment T used in this formula should be that coming upon the portion of the reverse shaft nearest the reversing engine. The reversing engine is usually placed near the middle of the length of the main engine, and it is generally safe to figure the shaft for the twisting moment necessary to move the low-pressure gear; for when these links make the greatest angle with the horizontal plane the other links are in a more advantageous position. In the preliminary design the bending moment can be neglected, and the working stress factor increased from 12 to 15 to allow for this neglect. After taking off the medium-pressure reverse-shaft levers, the size of the shaft running to the high-pressure levers can usually be decreased.

118. Valve Stem Load. — In using Formula (48), the weight of the valves, valve stems, crossheads, and blocks is usually obtained from data for similar engines. If no such data are available the loads coming upon the valve stems can be taken as 15 DE. D is the diameter of the valve in inches and E is the eccentricity in inches. In the case of twin valves the sum of the diameters should be used.

SECTION III

ENGINE BALANCING

119. Vertical Forces Balanced. — The unbalanced forces which are most objectionable in a marine engine are those which act in a vertical direction. The ship is usually much stiffer in a horizontal plane than it is in a vertical plane and as the horizontal disturbing forces with an upright engine are a good deal smaller than the vertical forces, attention is confined almost entirely to the vertical disturbing forces.

120. Motion of Parts. — The moving parts of an engine may be divided into four classes depending upon the character of the motion of the part. Some parts reciprocate, some rotate, some vibrate, and some both reciprocate and vibrate. The piston, piston rod, crosshead, slipper, valves, and valve stems have a reciprocating motion; the crank shaft, crank webs, crank pins, and eccentrics have a rotating motion; the drag rods and pump levers vibrate; the connecting rods and links both vibrate and reciprocate. The calculations for balancing the vertical forces would become too complicated if we attempted to treat the motion of each part with absolute correctness; so it is usual to group all parts into two classes, one rotating and the other reciprocating.

121. Division of Connecting Rod. — In order that this classification may be made it is necessary to assume that the connecting rod can be divided into two parts, one a reciprocating part acting at the crosshead and the other a rotating part acting at the crank pin.

Let M = mass of the rod.
l = distance between centers.
x = distance from crosshead to center of gravity of rod.

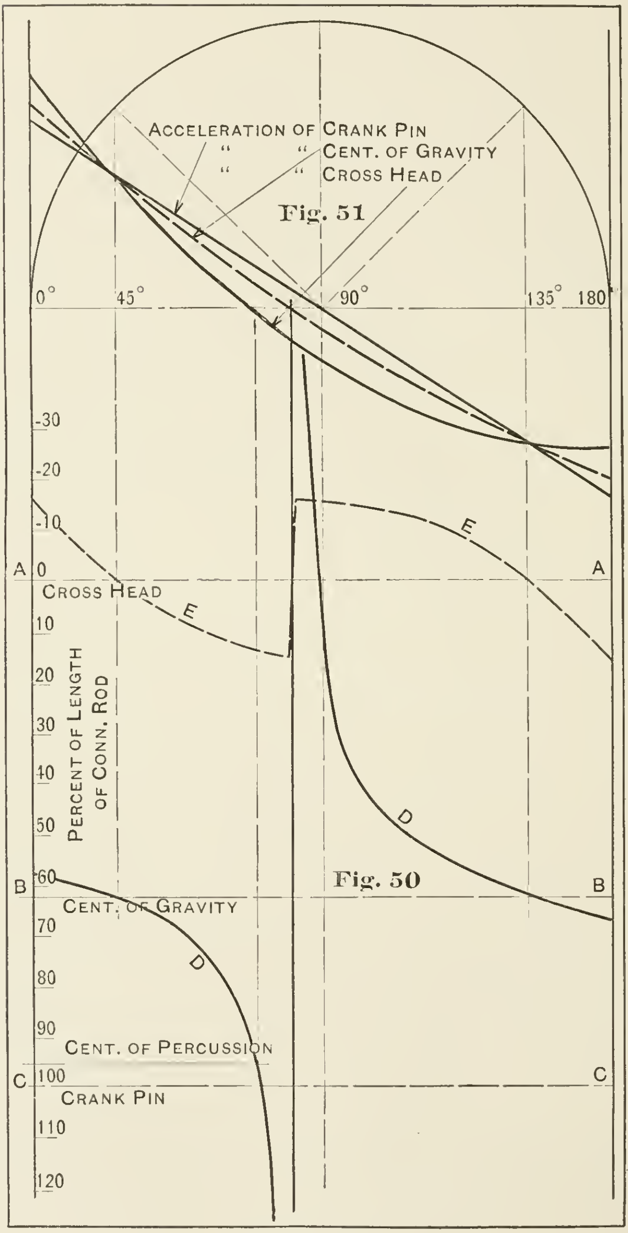
Then the mass acting at the crosshead is taken as $M\frac{l-x}{l}$, and the mass acting at the crank pin is $M\frac{x}{l}$. This is equivalent to assuming that the resultant of the vertical accelerating forces acts at the center of gravity of the rod throughout the revolution, or that the connecting rod is of infinite length. As a matter of fact the resultant of the vertical accelerating forces does not act at the center of gravity of the rod but is acting sometimes above and sometimes below that point at various parts of the stroke (see Fig. 50). A consideration of the velocities of the different parts of the rod in a vertical direction will show this to be true.

Professor Lanza gives the following formula for the distance from the crosshead to the point of application of the resultant of the vertical accelerating forces:

$$x_1 = \frac{f_1(l-p)+f_2p}{f_1(l-x)+f_2x}x. \tag{57}$$

f_1 = acceleration of crosshead.
f_2 = acceleration of crank pin in a vertical direction.
l = length of rod between centers.
p = distance from crosshead to center of percussion.
x = distance from crosshead to center of gravity.

122. Error in Division of Connecting Rod. — Fig. 50 is plotted by means of this formula on the supposition that the ratio of connecting rod to crank is 4, that $x = 0.62\,l$, and that $p = 0.96\,l$. The horizontal line BB shows the assumption made in regard to the point of application of the resultant of the accelerating forces and the curves D show the actual location of that point. In Fig. 51 curves are given which show the acceleration of the ends of the connecting rod and also the acceleration of its center of gravity. Where the acceleration of all three points is the same the rod has a motion of translation and the accelerating force is applied at the center of gravity of the rod. This occurs at crank angles of 45° and 135°. When the acceleration of the crosshead end of the rod is zero the accelerating force is applied at the center of percussion. When the acceleration of the center



Acceleration of Crank Pin
" " Cent. of Gravity
" " Cross Head
Fig. 51
0°
45°
90°
135°
180°
-30
-20
-10
A
0
Cross Head
E
10
20
30
40
50
60
B
Cent. of Gravity
70
80
90
Cent. of Percussion
C
100
Crank Pin
110
120
Percent of Length of Conn. Rod
D
E
D
Fig. 50
A
B
C

Figs. 50, 51.

of gravity of the rod is zero and when, consequently, the accelerating force is zero, the point of application is at infinity. Thus the maximum deviation from the assumed condition occurs at a time when the force itself is a minimum. The deviation of the assumed condition from the truth is shown by curve E plotted upon AA as a base. From 0° to 45° we have assumed more weight acting at the crank pin than we should, while from 45° to about 85° we have assumed less.

This assumption in regard to the portions of the connecting rod that shall be considered as acting at the crosshead and crank pin gives us the following rotating masses: — crank pin, crank webs, and a portion of the connecting rod, $M\frac{x}{l}$. The valve gears are also treated as rotating masses for reasons that will appear later. The reciprocating masses will be, — the piston, piston rod, crosshead, slipper, and the remainder of the connecting rod, $M\frac{l-x}{l}$.

123. Balance of Rotating Masses. — It is not at all difficult to balance the rotating masses, as it is only necessary to find the

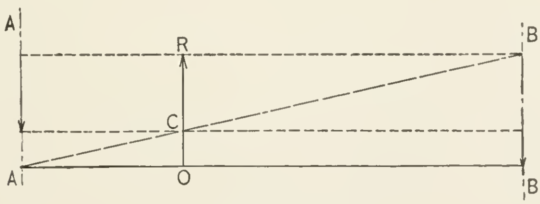


Fig. 52.

resultant of these masses and then to so place other rotating masses that this resultant shall be reduced to zero. It is usual to do this by placing one or more weights on the crank webs at either end of the engine. The resultant can be found by graphical means, as shown in Fig. 52. Let the unbalanced mass be represented in magnitude by OR and let the balance weights be placed in the planes AA and BB. Project the length OR upon the plane BB and draw AB. This diagonal line will divide OR into two parts, such that, if OC is placed in the plane BB, and CR

in the plane AA, acting in a direction opposite to OR, the moment of OC about O will be equal to the moment of CR about O, and we shall have OR balanced for hammering and tilting. If the mass to be balanced is not between the two balance planes the construction shown in Fig. 53 can be used. In this case the

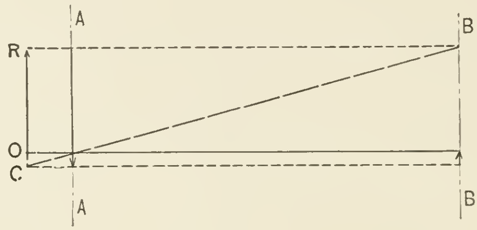


Fig. 53.

mass acting at AA is greater than the mass at O and acts in the opposite direction, while the mass at BB is equal to the difference of these two and acts in the same direction as OR.

This method can be applied to an engine with any number of cylinders. In Fig. 54 it is applied to a three-crank engine with

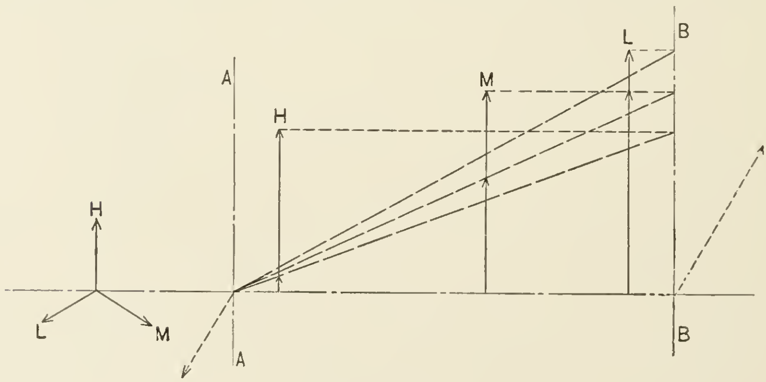


Fig. 54.

cranks at 120°. In each balance plane there will be three components to be combined at the proper angles, and the balance mass must be equal to the resultant and act in an opposite direction. See Figs. 55 and 56.

When the unbalanced masses are not rotating on a radius

equal to the length of the crank arm it is usual to reduce them to equivalent masses acting at this radius by means of the following:

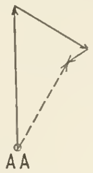


FIG. 55.

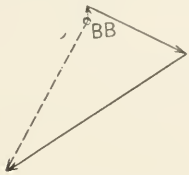


FIG. 56.

equivalent mass × crank arm length = actual mass × arm of center of gravity. In the case of the crank webs a certain portion is balanced because of its construction and only the unbalanced portion need be considered.

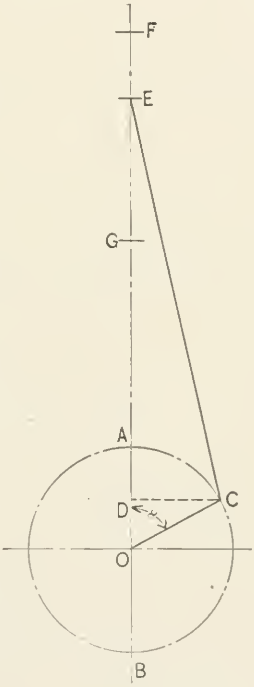


FIG. 57.

124. Acceleration of Crosshead. — The rotating masses will be balanced not only for forces acting in a vertical plane but also for forces in a horizontal plane. When we come to the reciprocating masses, however, we have to deal with forces which act in a vertical plane only. It is usual to consider that the speed of rotation of the crank shaft is uniform. The end of the connecting rod which is attached to the crank pin will have an acceleration in a vertical direction due to harmonic motion while that of the crosshead end will differ from this because of the angularity of the connecting rod. In Fig. 57, $AD = s$ = space passed over by rotating parts in a vertical direction in time t. Angular velocity $= \alpha$.

$$s = r - r \cos \alpha t,$$

$$\text{velocity} = v_2 = \frac{ds}{dt} = \alpha r \sin \alpha t,$$

$$\text{acceleration} = f_2 = \frac{d^2s}{dt^2} = \alpha^2 r \cos \alpha t. \qquad (58)$$

In the case of the crosshead:

$FE = s =$ space passed over by reciprocating parts in a vertical direction in time t.

$$s = FA + AO - OD - ED = l + r - r \cos \alpha t - \sqrt{l^2 - r^2 \sin^2 \alpha t}$$

$$= r\left(1 + \frac{l}{r} - \cos \alpha t - \sqrt{\left(\frac{l}{r}\right)^2 - \sin^2 \alpha t}\right).$$

$$v_1 = \frac{ds}{dt} = \alpha r \sin \alpha t \left\{1 + \frac{\cos \alpha t}{\sqrt{\left(\frac{l}{r}\right)^2 - \sin^2 \alpha t}}\right\}.$$

$$f_1 = \frac{d^2 s}{dt^2} = \alpha^2 r \left\{\cos \alpha t + \frac{\cos 2 \alpha t}{\sqrt{\left(\frac{l}{r}\right)^2 - \sin^2 \alpha t}} + \frac{\sin^2 2 \alpha t}{4\left[\left(\frac{l}{r}\right)^2 - \sin^2 \alpha t\right]^{\frac{3}{2}}}\right\}. \quad (59)$$

It is customary to neglect the last term of this expression since its maximum value when $\frac{l}{r} = 4$ is 0.004, and the denominator of the preceding term is also simplified by dropping the *sin*^{2} αt.

The approximate expression for f_1 then becomes

$$f_1 = \alpha^2 r \left(\cos \alpha t + \frac{r}{l} \cos 2 \alpha t\right).$$

$$\alpha r = \text{linear velocity of crank pin} = v.$$

$$\therefore \quad f_1 = \frac{v^2}{r}\left(\cos \alpha t + \frac{r}{l} \cos 2 \alpha t\right). \quad (60)$$

If M = mass of the reciprocating parts the accelerating force is

$$F = \frac{Mv^2}{r}\left(\cos \alpha t + \frac{r}{l} \cos 2 \alpha t\right). \quad (61)$$

Another form in which this accelerating force can be expressed is as follows:

$$F = \frac{Mv^2}{r}\left\{\cos \alpha t \right.$$
$$+ \cos 2 \alpha t \left[\frac{r}{l} + 4\left(\frac{r}{l}\right)^3 + \frac{15}{128}\left(\frac{r}{l}\right)^5 + \cdots\right]$$
$$+ \cos 4 \alpha t \left[\frac{1}{4}\left(\frac{r}{l}\right)^3 + \frac{3}{16}\left(\frac{r}{l}\right)^5 + \cdots\right]$$
$$\left. + \cos 6 \alpha t [\cdots]\right\}. \quad (62)$$

etc.

Approximately,

$$
\begin{aligned}
F = {} & M\frac{v^2}{r}\cos \alpha t && (A)\\
& + M\frac{v^2}{r}\frac{r}{l}\cos 2\,\alpha t && (B)\\
& + M\frac{v^2}{r}\frac{1}{4}\left(\frac{r}{l}\right)^3\cos 4\,\alpha t. && (C) \qquad (63)\\
& \text{etc.}
\end{aligned}
$$

Since $v = 2\,\pi r n.$

$$
\begin{aligned}
(A) &= M\,4\,\pi^2 n^2 r \cos \alpha t.\\
(B) &= M\,4\,\pi^2\,(2\,n)^2\frac{r^2}{4\,l}\cos 2\,\alpha t.\\
(C) &= M\,4\,\pi^2\,(4\,n)^2\frac{r^4}{64\,l^3}\cos 4\,\alpha t.\\
& \text{etc.}
\end{aligned}
$$

These last three expressions can be represented graphically as shown in Fig. 58.

Let r = radius of crank = 1.
$l = 4.$

M is the mass of the piston, piston rod, crosshead, slipper, and upper end of the connecting rod.

125. Primary and Secondary Masses. — The mass M rotating with an angular velocity α on a radius $r = 1$ would give rise to a force whose vertical component is called the *primary* hammering force. The mass M rotating on a radius $\frac{r^2}{4\,l} = \frac{1}{16}$, with an angular velocity $2\,\alpha$, gives rise to a force whose vertical component is called the *secondary* hammering force. These are the only two forces usually considered, but if a nearer approximation were desired we could add a third mass M rotating on a radius $\frac{r^4}{64\,l^3} = \frac{1}{4096}$ with an angular velocity $4\,\alpha$.

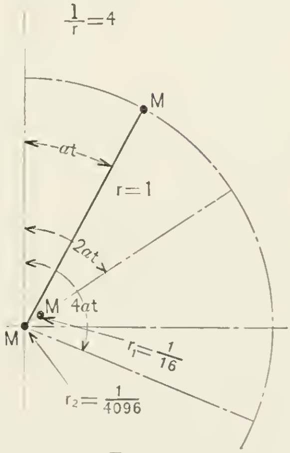


Fig. 58.

126. Approximations. — We have made, then, two principal approximations: first, that the length of the connecting rod is infinite and that its mass can be divided into two fixed portions, one acting at the crosshead and the other at the crank pin; second, that the effect of the reciprocating masses is essentially the same as if we were dealing with the vertical components of the centrifugal forces arising from the rotation of two masses, each equal to the reciprocating masses, — one making the same number of revolutions as the main engine on a radius r, the other revolving twice as fast on a radius $\frac{r^2}{4l}$.

127. Valve Gear Treated as Rotating Mass. — It will be seen that the expression for the secondary force contains the expression $\frac{r^2}{4l}$, so that when the ratio of l to r is large the value of the secondary force is small. In the case of the valve gear the length of the eccentric rod is usually from 20 to 30 times the eccentricity, so that we can neglect the secondary forces without any great error in this case. This makes the disturbance due to the valve gear equivalent to the vertical component of the centrifugal force arising from the uniform rotation of a mass equal to the mass of the valve gear. If we balance this with a rotating mass we shall have overbalance in a horizontal direction since the valve gear has motion only in a vertical direction.

It is customary to find the resultant effect of all the valve gears and then balance this resultant with a rotating mass. In the case of the M.P. cylinder or cylinders it is possible to have the valves take steam on the inside or on the outside and the resultant unbalanced force may be reduced by changing the angle of the eccentrics to suit one method or the other. The engine is balanced for the go-ahead position with the ahead eccentric actuating the valve, valve stem, crosshead, half the drag rods, half the link, one eccentric rod, and one strap. The backing eccentric actuates half the link, one eccentric rod, and one strap. The resultant for the valve gears can be found by the method shown in Figs. 52, 53, and 54.

By means of these assumptions and approximations the

question of engine balancing is reduced to the consideration of three revolving masses for each cylinder. One of these is the sum of the masses which are supposed to have a motion of rotation; namely, the unbalanced portions of the crank webs, crank pins, lower part of connecting rod, eccentrics, eccentric straps, eccentric rods, links, link blocks, valve stems, and valves, all reduced to equivalent masses acting at the crank pin and rotating at the same number of revolutions as the main engine. Another of these masses, called the primary mass, is the sum of those which are supposed to reciprocate and is made up of the upper part of the connecting rod, crosshead, slipper, piston rod, and piston, all rotating with the same number of revolutions as the main engine, upon a radius equal to the length of the crank arm. The third mass is called the secondary mass, and is equal to the primary mass, but is rotating with twice the number of revolutions of the main engine, upon a radius equal to $\frac{r^2}{4\,l}$.

128. Balance with Bob Weights. — The reciprocating masses are not as easily balanced as the rotating, by reason of the fact that we have to deal with the secondary masses. The primary masses might be balanced by a bob weight actuated by a crank making the same number of revolutions as the main engine. The secondary masses would have to be balanced by a bob weight actuated by a crank making twice as many revolutions as the main engine, and this would necessitate the introduction of gearing which is usually not practicable. If bob weights are to be used we would proceed in the same manner as for the balance of rotating masses: select the planes in which the bob weights are to act, find the components which are to act in those planes, and combine the components at the proper angle with one another to find the resultant weight and the angle at which the actuating crank is to be set.

129. Balance without Use of Extra Weights. — It is usual, however, to avoid the use of any extra cranks and bob weights, and to arrange the crank angles, relative location of cylinders, and weights of the reciprocating parts so as to secure as nearly as possible perfect balance. If we have a force diagram such as

is shown in broken lines in Fig. 59 for the primary forces of a four-cylinder engine we should need a force *Od* to close the diagram, but if we arrange the crank angles differently and perhaps increase or decrease some of the reciprocating masses we can get the diagram to close as shown in the full lines of Fig. 59. It might be thought that since we are in reality concerned only with the vertical components of these forces, if we had a

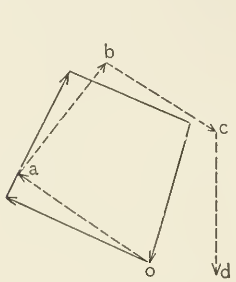


FIG. 59.

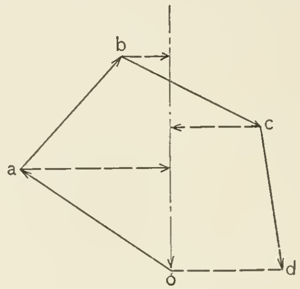


FIG. 60.

condition such as is shown by Fig. 60 the engine would be balanced. The engine would be balanced for that crank position but as soon as the cranks revolve slightly the balance would be destroyed. In order that the arrangement of cranks and weights shall be such as to be balanced in all positions it is necessary that the sum of both the horizontal and vertical components of the forces shall equal zero.

130. Equations for Force and Moment Diagrams. — The determination of the conditions for perfect balance will involve the use of two equations for each force or moment diagram. We shall have one force diagram for the primary hammering forces, another for the secondary hammering forces, one moment diagram for the primary tilting couples, and another for the secondary tilting couples. In all we shall have to deal with eight equations, each equation involving as many terms as there are cranks.

Primary forces disappear if

$$(1) \quad \frac{v^2}{r}(M_1 \cos A + M_2 \cos B + M_3 \cos C + \cdots) = 0.$$

$$(2) \quad \frac{v^2}{r}(M_1 \sin A + M_2 \sin B + M_3 \sin C + \cdots) = 0.$$

Primary couples disappear if

(3) $\frac{v^2}{r}(M_1a_1 \cos A + M_2a_2 \cos B + M_3a_3 \cos C + \cdots) = 0.$

(4) $\frac{v^2}{r}(M_1a_1 \sin A + M_2a_2 \sin B + M_3a_3 \sin C + \cdots) = 0.$

Secondary forces disappear if

(5) $\frac{r}{l}\frac{v^2}{r}(M_1 \cos 2A + M_2 \cos 2B + M_3 \cos 2C + \cdots) = 0.$

(6) $\frac{r}{l}\frac{v^2}{r}(M_1 \sin 2A + M_2 \sin 2B + M_3 \sin 2C + \cdots) = 0.$

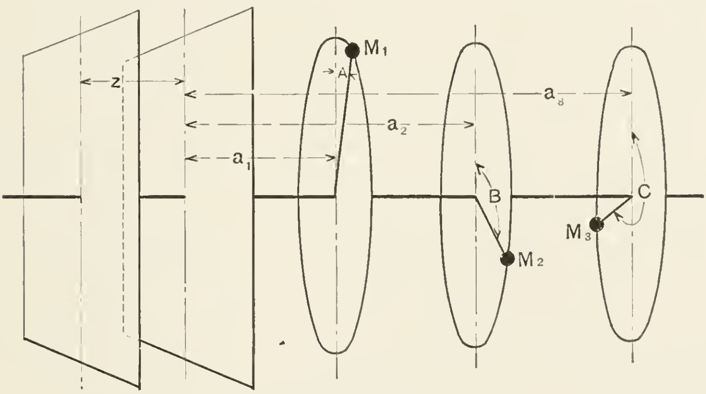


FIG. 61.

Secondary couples disappear if

(7) $\frac{r}{l}\frac{v^2}{r}(M_1a_1 \cos 2A + M_2a_2 \cos 2B + M_3a_3 \cos 2C + \cdots) = 0.$

(8) $\frac{r}{l}\frac{v^2}{r}(M_1a_1 \sin 2A + M_2a_2 \sin 2B + M_3a_3 \sin 2C + \ldots) = 0.$

v = linear velocity of crank pin.

r = radius of crank-pin circle.

l = length of connecting rod between centers.

M_1, M_2, M_3, etc. = masses of reciprocating parts of different cylinders.

a_1, a_2, a_3, etc. = distance of different cylinders from reference plane.

A, B, C, etc. = angles that cranks make with reference line (see Fig. 61).

131. Order in which Equations Must be Used. — It is obvious that the equations must be taken in pairs since satisfying equation (1) but not (2) would mean that for one particular position of the cranks the sum of the vertical components equals zero, but if the cranks are revolved slightly equation (1) will not equal zero.

In the above equations we have the quantities a_1, a_2, a_3, etc., which are the distances from some chosen reference plane to the planes in which the reciprocating masses are acting. It is possible that we may so choose the location of our reference plane that equations (3), (4), (7), and (8) will be satisfied, but if a new reference plane is used these equations may not be satisfied. If the engine is "perfectly" balanced the moment diagrams ought to close no matter where the reference plane is taken.

In Fig. 61 let the reference plane be moved a distance z. Equation (3) will then become

$$\frac{v^2}{r}[M_1(a_1+z)\cos A + M_2(a_2+z)\cos B + M_3(a_3+z)\cos C + \ldots] = 0,$$

or,

$$\frac{v^2}{r}[(M_1a_1\cos A + M_2a_2\cos B + M_3a_3\cos C + \ldots) + z(M_1\cos A + M_2\cos B + M_3\cos C + \ldots)] = 0.$$

This is equation (3) with the expression $z(M_1 \cos A + M_2 \cos B + M_3 \cos C)$ added, and will still be equal to zero if the quantity $(M_1 \cos A + M_2 \cos B + M_3 \cos C)$ is equal to zero. This latter expression is the same as equation (1). In other words, we must satisfy equations (1) and (2) before we satisfy (3) and (4) and must satisfy (5) and (6) before satisfying (7) and (8). Not only must the equations be taken in pairs but they must be taken in a certain sequence. If we have enough unknowns to satisfy only two equations we could choose (1) and (2), or (5) and (6), but since the primary forces are much larger than the secondary, we would naturally choose (1) and (2). If there are enough unknowns for four equations we can satisfy (1), (2), (3), and (4), or (1), (2), (5), and (6). It would of course

be possible to satisfy (5), (6), (7), and (8), but it would not be wise to eliminate the lesser forces instead of the greater. If six equations could be satisfied the choice would probably be (1), (2), (3), (4), (5), and (6).

132. Number of Unknown Quantities. — The number of unknowns in any case will be a function of the number of cranks. In order that we may be able to make use of the results obtained from our force and moment diagrams we must know one mass and the distance of that mass from our reference plane. If n is the number of cranks, the greatest possible number of unknown masses M will be $(n - 1)$, and the unknown arms a can also be as great as $(n - 1)$. The greatest possible number of unknown angles between the cranks will be $(n - 1)$. Therefore the greatest number of unknown quantities in any case will be $3(n - 1)$.

133. Single-crank engine. — In this case $3(1 - 1) = 0$. None of the equations can be satisfied, so the reciprocating parts cannot be balanced. The rotating parts can be balanced by other rotating weights.

134. Two-crank engine. — In this case the greatest number of unknowns can be $3(2 - 1) = 3$. Since the equations have to be taken in pairs we can satisfy only two, and the equations chosen will probably be (1) and (2).

$$\frac{v^2}{r}(M_1 \cos A + M_2 \cos B) = 0.$$

$$\frac{v^2}{r}(M_1 \sin A + M_2 \sin B) = 0.$$

Let M_1 be known, and the position of the cranks such that $A = 0°$.

$$\frac{v^2}{r}(M_1 + M_2 \cos B) = 0.$$

$$\frac{v^2}{r}(M_2 \sin B) = 0.$$

M_2 cannot $= 0$, $\therefore$ $\sin B = 0$, and B must $= 0°$ or $180°$. If $B = 0°$, we have a one-crank engine. $\therefore$ B must $= 180°$. If $B = 180°$, $M_1 = M_2$. We have as a result of these conditions

two cylinders in the same plane, the reciprocating masses equal and actuated by cranks at 180°. This gives the arrangement recommended by McAlpine. (See Fig. 62.)

135. Three-crank Engine. — The greatest number of unknowns for this engine will be $3(3-1)=6$. When the first six equations are solved the following relations are obtained:

$$M_1 = M_2 = M_3.$$
$$\cos B = \cos C = -\tfrac{1}{2}.$$
$$\sin B = -\sin C.$$
$$a_1 = a_2 = a_3.$$

This gives an engine with the reciprocating masses of the three cylinders all equal; the cranks at 120°, and all in the same plane

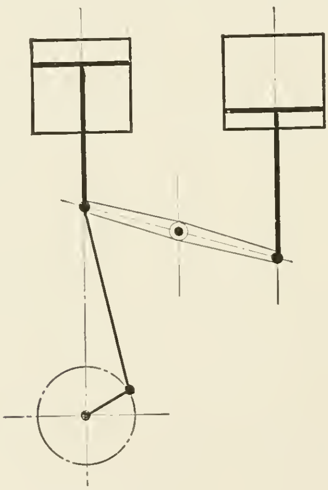


Fig. 62.

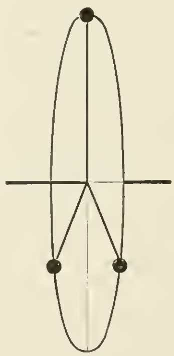


Fig. 63.

(see Fig. 63). This arrangement is of no practical use. If we give a_2 and a_3 such values as will separate the cylinders we shall still have four unknowns and can satisfy the first four equations. The results of this solution are as follows:

$$M_1 + M_3 = M_2.$$
$$M_2a_2 - M_3a_3 = 0.$$
$$\cos B = -1.$$
$$\cos C = +1.$$

This gives the arrangement shown in Fig. 64; one heavy set of reciprocating masses between two lighter sets and making an angle of 180° with them. It is not desirable to have the cranks at 180° on account of the great variation in turning moments which results from this arrangement. Engines are sometimes made with the low-pressure cylinder in the middle but with the cranks at 120°.

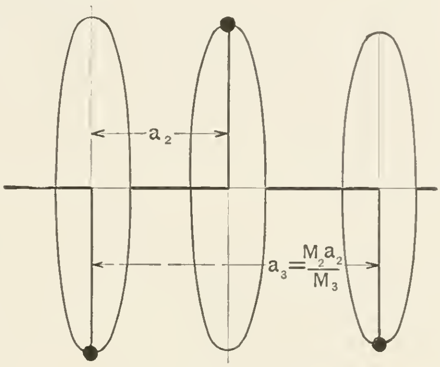


Fig. 64.

136. Four-crank Engine. — The total unknowns with this number of cranks can be 3 (4 − 1) = 9. As there are only eight equations to solve we will let a_2 be a known quantity as well as a_1. When the eight equations are solved we find that the following conditions must exist for "perfect" balance:

$$M_2 = M_3.$$
$$M_4 + M_1 = M_2 = M_3.$$
$$M_4 a_4 = M_2 a_2.$$
$$a_3 = a_2.$$
$$\sin C = -\sin B.$$
$$\cos C = \cos B = -\tfrac{1}{2}.$$
$$\cos D = 1.$$
$$\sin D = 0.$$

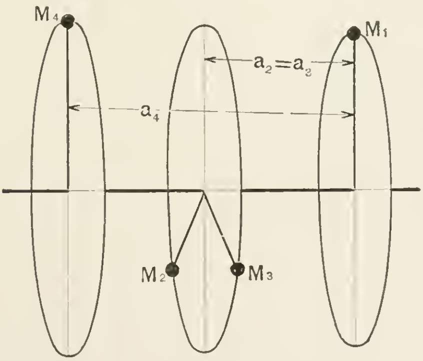


Fig. 65.

This results in the arrangement shown in Fig. 65, an arrangement which is not possible since it brings two cylinders in the same transverse plane. If we increase our known quantities by two and reduce our unknowns to six we can satisfy the first six equations and get a possible solution.

137. Yarrow-Schlick-Tweedy System. — The two additional known conditions can be so chosen that we get an engine with a

symmetrical arrangement of cylinders. This arrangement is commonly known as the Yarrow-Schlick-Tweedy system of balancing. We will use the notation common to this system, see Fig. 66, in which α is the angle between cranks 1 and 4, γ is the angle between cranks 2 and 3, β is the angle between cranks 1 and 3, and β_1 is the angle between cranks 2 and 4.

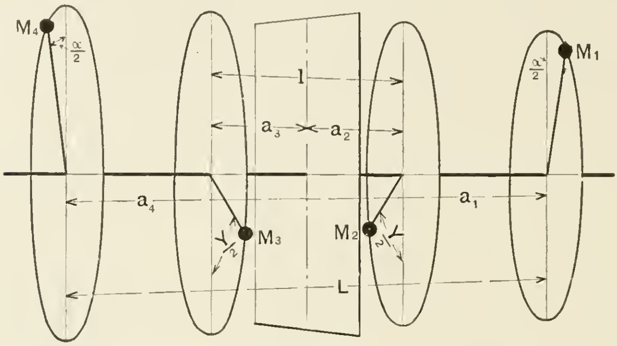


FIG. 66.

M_1 and a_1 must be known anyway and the three additional known conditions will be as follows:

Let

$$a_4 = -a_1.$$
$$a_2 = -a_3.$$
$$M_4 = M_1.$$

This gives the symmetrical arrangement shown in Fig. 66. When the first six equations are solved we have the following results:

$$M_2 = M_3.$$
$$\beta = \beta_1.$$
$$M_1 L \sin\frac{\alpha}{2} = M_3 l \sin\frac{\gamma}{2}.$$
$$\cos\frac{\alpha}{2}\cos\frac{\gamma}{2} = \frac{1}{2}.$$
$$L \tan\frac{\alpha}{2} = l \tan\frac{\gamma}{2}.$$
$$M_3 \cos\frac{\gamma}{2} = M_1 \cos\frac{\alpha}{2}.$$

On account of the symmetry of the arrangement we can reduce the actual unknown relations to three, as follows:

$$M_3 \cos\frac{\gamma}{2} = M_1 \cos\frac{\alpha}{2}. \tag{64}$$

$$\cos\frac{\gamma + \alpha}{2} = \frac{1}{2} + h - \sqrt{h^2 + \frac{3}{4}}. \tag{65}$$

$$\cos\frac{\gamma - \alpha}{2} = \frac{1}{2} - h + \sqrt{h^2 + \frac{3}{4}}. \tag{66}$$

$$h = \frac{1}{4}\left(\frac{L}{l} + \frac{l}{L}\right). \tag{67}$$

This is the system of balancing most commonly used and the more nearly the value of $\frac{l}{L}$ approaches unity the more nearly the angles α and γ approach 90°. It is difficult, however, to get the value of $\frac{l}{L}$ to exceed 0.5 without lengthening the engine considerably.

138. Unsymmetrical Four-crank Arrangement. — In addition to the above symmetrical arrangements there are an infinite number of unsymmetrical arrangements of cylinders, crank angles, and reciprocating weights which will give an engine balanced for primary forces, secondary forces, and primary tilting couples.

Mr. Chas. E. Inglis, M. A., in a paper read before the Institute of Naval Architects in 1911, gives certain diagrams which greatly reduce the labor involved in selecting the proper relations for balance. Usually the relative location of cylinders will be quite closely determined. There are certain practical limits set upon the maximum length of an engine, and the sizes of the different cylinders and valve chests, and the sizes of the crank-shaft parts will determine the minimum distance between cylinder centers. Fig. 67, which is taken from the above-mentioned paper, is so drawn that if the relative location of the cylinders is known the crank angles which will give freedom from primary tilting couples can be determined. These angles should satisfy the following equation:

$$2 \cos\frac{\alpha}{2} \cos\frac{\gamma}{2} = \cos\frac{\beta - \delta}{2}. \tag{68}$$

In order that freedom from primary and secondary hammering may be obtained the following relation between masses and crank angles should exist:

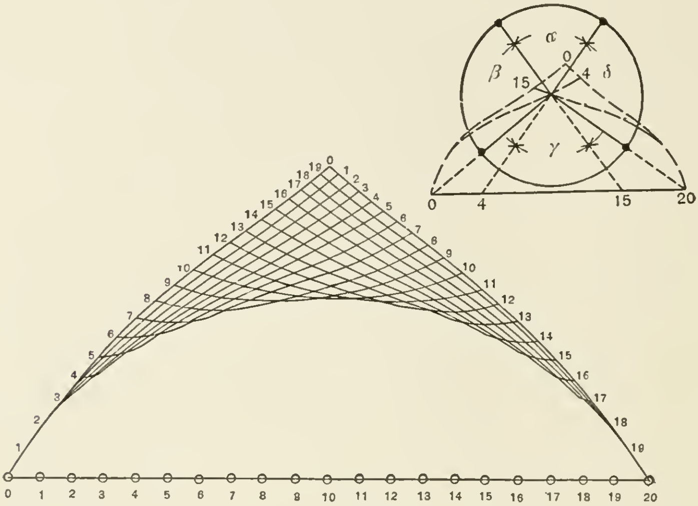


Fig. 67.

$$M_2 = M_0 \frac{\sin \alpha \sin \frac{\delta}{2} \sin \left(\alpha + \frac{\delta}{2}\right)}{\sin \beta \sin \frac{\gamma}{2} \sin \left(\beta + \frac{\gamma}{2}\right)}. \qquad (69)$$

$$M_3 = M_0 \frac{\sin \alpha \sin \left(\frac{\alpha + \beta}{2}\right) \sin \left(\frac{\alpha - \beta}{2}\right)}{\sin (\beta + \gamma) \sin \left(\beta + \frac{\gamma}{2}\right) \sin \frac{\gamma}{2}}. \qquad (70)$$

$$M_1 = M_0 \frac{\sin \delta \sin \frac{\delta + \gamma}{2} \sin \left(\frac{\delta - \gamma}{2}\right)}{\sin (\beta + \gamma) \sin \left(\gamma + \frac{\beta}{2}\right) \sin \frac{\beta}{2}}. \qquad (71)$$

M_0 can be taken as unity, in which case the equations will give the relative weights of the masses.

The disturbance due to the secondary tilting couples can be determined by introducing the above distances, angles, and masses into equations (7) and (8).

The significance of Fig. 67 can be shown by the following proof. In Fig. 68 let ce represent by its length the mass of M_1 in Fig. 69, and let its direction be parallel to the crank arm of M_1. In the same way let ea represent M_3, ag represent M_2, and gc represent M_4. Since these four lines make a closed diagram the resultant of the primary hammering forces is zero and we can investigate the primary tilting couples. If we take our plane of reference through M_1 the polygon of primary tilting couples will be a triangle, such as efa in Fig. 68. Draw def parallel to ag, af parallel to cg, and eb parallel to cg. If the resultant of the primary tilting couples is to be zero

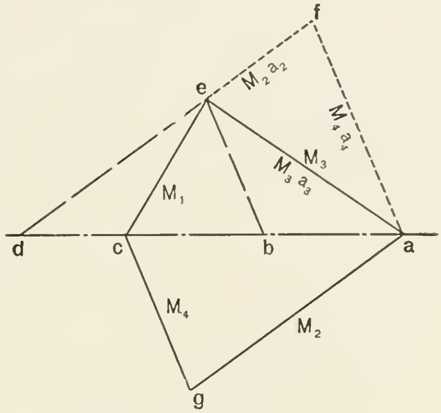


FIG. 68.

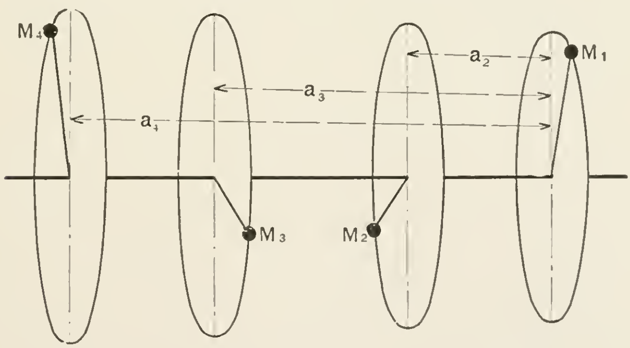


FIG. 69.

M_2a_2, M_3a_3, and M_4a_4 must have such values that they can be represented by ef, ea, and af.

$$M_2a_2 : M_3a_3 : M_4a_4 = ef : ea : af,$$

then

$$\frac{M_2a_2}{M_4a_4} = \frac{ef}{af};$$

but $\quad \dfrac{M_2}{M_4} = \dfrac{ag}{cg} = \dfrac{df}{af}, \qquad \therefore \quad \dfrac{a_2}{a_4} = \dfrac{ef}{af} \times \dfrac{af}{df} = \dfrac{ef}{df} = \dfrac{ab}{ad};$

also, $\quad \dfrac{M_3 a_3}{M_4 a_4} = \dfrac{ae}{af},$

but $\quad \dfrac{M_3}{M_4} = \dfrac{ae}{cg}, \qquad \therefore \quad \dfrac{a_3}{a_4} = \dfrac{ae}{af} \times \dfrac{cg}{ae} = \dfrac{cg}{af} = \dfrac{ac}{ad}.$

This proves that the resultant of the primary tilting couples will be zero if $a_2 : a_3 : a_4 = ab : ac : ad$. The point e of any such

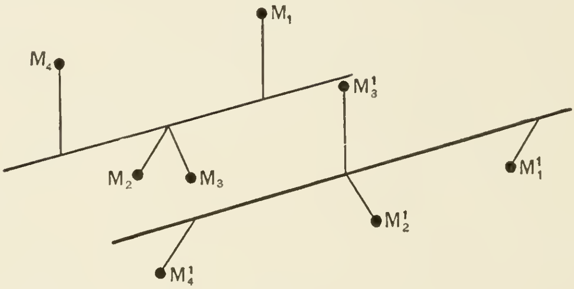


FIG. 70.

diagram as shown in Fig. 68 can be obtained from Fig. 67 by finding the intersection of the curves corresponding to points b and c.

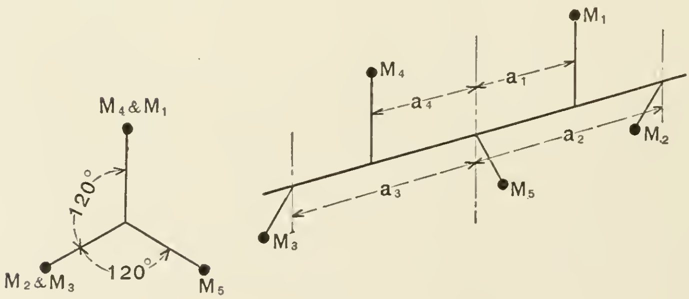


FIG. 71.

139. Engines with Five or more Cranks. — The arrangement which will give balance in engines with five or more cranks can be determined by combining two or more groups of "perfectly" balanced four-crank engines. In Fig. 71 we see how the two sets of four-crank engines shown in Fig. 70 can be combined to

give a five-crank engine. The two sets of four-crank engines are identical except that the distance between the outside cylinders is greater in one case than in the other. If one engine is imposed upon the other so that M_2, M_3, and M_2', M_3' are in the same plane we shall have three masses in this plane which fulfill the requirements for a "perfectly" balanced three-crank engine (see Fig. 63); i.e., the three masses are equal and the cranks make angles of 120°. This balanced set of three cranks can be eliminated and the remaining five cranks will be in "perfect" balance. The conditions will be as follows:

$$M_1 + M_4 = M_2 + M_3 = M_5.$$
$$M_1a_1 = M_4a_4.$$
$$M_3a_3 = M_2a_2.$$

M_1 and M_4 are in the same axial plane.
M_2 and M_3 are in the same axial plane.
Cranks are in planes making 120°.

In the same way three sets of "perfectly" balanced four-crank engines can be combined to give a six-crank engine after the

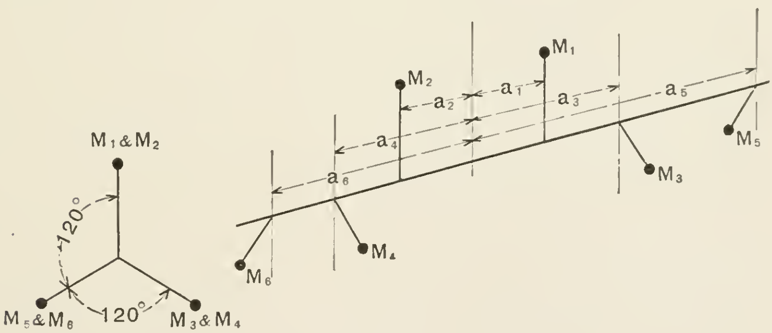


FIG. 72.

elimination of two sets of balanced three-crank engines. Fig. 72 shows the resulting arrangement for the six-crank engine. The conditions will be as follows:

$$M_1 + M_2 = M_3 + M_4 = M_5 + M_6.$$
$$M_1a_1 = M_2a_2.$$
$$M_3a_3 = M_4a_4.$$
$$M_5a_5 = M_6a_6.$$

M_1 and M_2 are in the same axial plane.
M_3 and M_4 are in the same axial plane.
M_5 and M_6 are in the same axial plane.
The axial planes make angles of 120°.

140. Summary. — In the case of three-crank engines the investigation shows that the primary and secondary hammering forces can be balanced if the heaviest cylinder is placed in the middle of the engine and the cranks are 180° apart. The nearest approach to this in practice is to place the L.P. cylinder between the H.P. and the M.P., but the cranks are placed 120° apart to keep the turning moment more nearly constant.

Four-crank engines can be balanced for primary and secondary hammering forces and for the primary tilting couples. The most common form in practice is that used in the case of four-cylinder Triples, where the L.P. cylinders are placed on the ends of the engine with the H.P. and M.P. between them.

Five- and six-crank engines can be balanced for primary and secondary hammering forces and primary and secondary tilting couples. Five-crank engines are seldom built, and six-crank engines are rarely found except in small speed boats and automobiles. Even in five- and six-crank engines, however, the balance is not absolutely perfect because of the assumptions made regarding the connecting rod and because of the simplified expression used for the acceleration of the crosshead.

SECTION IV

CONDENSERS AND AIR PUMPS

141. Partial Pressures. — Before taking up the effect of air upon condensation, attention must be directed to Dalton's law of partial pressure for mixed gases. If two gases, such as air and water vapor, are mixed in a receptacle, the total pressure exerted is the sum of the pressures that each gas would exert if the same weight of gas which is present in the mixture occupied the entire volume of the receptacle alone. Thus, if we should take 1 cubic foot of saturated water vapor at 102° F. and force it into a receptacle containing a cubic foot of air at 102° F. and at a pressure of 0.1 pound absolute, the total pressure of the cubic foot of mixture would be 1.1 pounds since water vapor at 102° F. has a pressure of 1 pound absolute. This is true only, however, when we have an equilibrium of temperature. If we should take a cubic foot of steam at 120° F. and add it to a cubic foot of air at 70° F. and 0.1 pound absolute the total pressure would depend upon the final temperature assumed by the mixture of steam, water, and air.

142. Effect of Air upon Rate of Condensation. — A paper was read before the Victorian Institute of Engineers on Dec. 6, 1905, by James Alexander Smith, in which the effect of air upon surface condensation was discussed. The experiments were made upon a small scale and under conditions differing somewhat from those existing in surface condensers as ordinarily used. The experiments were conducted by means of a cylinder $7\frac{1}{2}$ inches in diameter and 3 feet long. Two $\frac{5}{8}$-inch tubes 3 feet long, connected in series, ran through the condenser and carried the cooling water. Burners were placed under the tank and were used to generate steam and keep the conditions uniform while condensation took place on the surface of the tubes. The receptacle served as boiler and condenser combined. The ex-

perimental results are valuable, however, as they show very clearly the injurious effect of air in condensers. Fig. 73, taken from this paper, shows the effect of different air mixtures upon the rate of heat transmission. When the partial pressure of the air is zero we are dealing with saturated steam and it will be

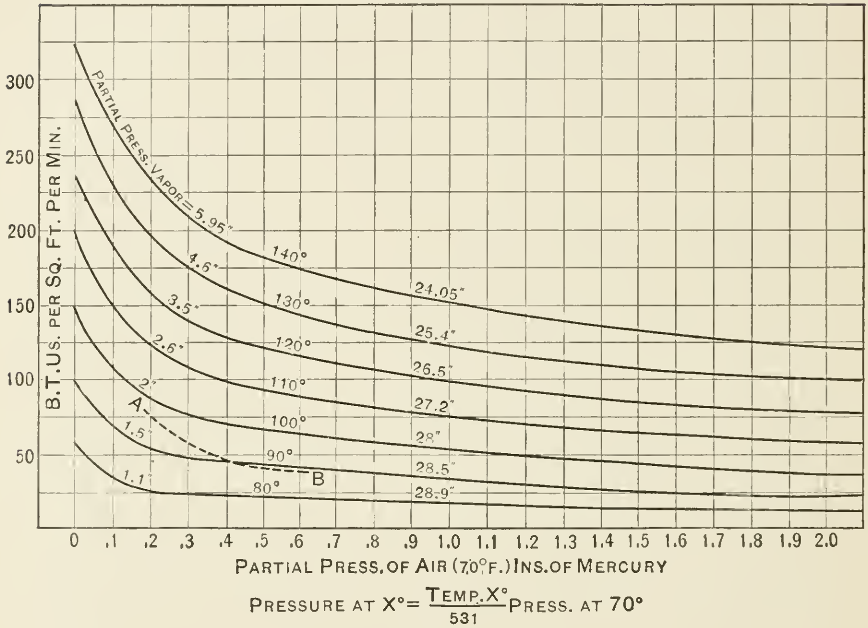


FIG. 73.

noticed that a small addition of air causes a rapid decrease in the rate of condensation especially at high vacua. A partial air pressure of 0.2 inch at a vapor temperature of 80° F. causes the efficiency of the surface to be decreased by more than 50 per cent. The same partial pressure at a vapor temperature of 140° F. causes a smaller loss of about 27 per cent in surface efficiency, although the decrease in B.t.u. transmitted is larger than in the first case.

In the experimental condenser all the temperatures of vapor, air, and water were in equilibrium and the air was uniformly distributed throughout the condenser. In actual practice we do not have this equilibrium. In the steam entering the top of the

condenser the proportion of air is so small that the steam is practically saturated and the temperature is that corresponding to the pressure of saturated steam. As the mixture of steam and air passes between the tubes the steam is condensed and the proportion of steam to air becomes less so that the partial pressure of the air increases and the temperature falls. We have, then, practically saturated steam at the top of the condenser and almost pure air at the bottom with a varying mixture in between. There is probably a tendency for the air to accumulate around the tubes, since there is a movement of the steam towards the cooling surface where it is condensed, leaving the air uncondensed. The velocity of steam and air passing over the tubes will prevent this accumulation of air to any great extent. The time taken by the steam and air to pass through the condenser is only a part of a second due to the fact that the velocity between the tubes is usually from 100 to 200 feet per second at the top, and about 20 feet per second in the pipe leading to the air pump.

By reason of this variation in the temperature of the mixture as it passes through the condenser, the rate of heat transmission would not follow any one of the isothermals shown in Fig. 73 but would be a steeper curve, such as AB, cutting across the isothermals.

It should be noticed in Fig. 73 that the partial pressures are given for air at 70° F. If it is desired to find the pressure of the mixture of air and vapor which would transmit 50 B.t.u. per square foot per minute at a temperature of 90°, we should take the partial pressure of the steam, 1.5 inches of mercury, and add to it $0.24 \times \frac{551}{531} = 0.249$ inch.

The total pressure would then be $1.5 + 0.249 = 1.749$ inches mercury, or about 0.875 pound absolute.

The effect of air upon the rate of temperature increase of water passing through the condenser tube is shown by the curves of Fig. 74. Curves A and B are from Mr. Smith's experiments, A is for saturated steam and B is for a mixture of steam and air of such proportion that the partial air pressure is 0.22 inch of

mercury. In the experimental condenser everything was in equilibrium and there was no movement of the steam except as new steam was generated to take the place of the steam condensed. It is probable that there was a tendency for the air to accumulate around the tubes. The other curves in Fig. 74 are

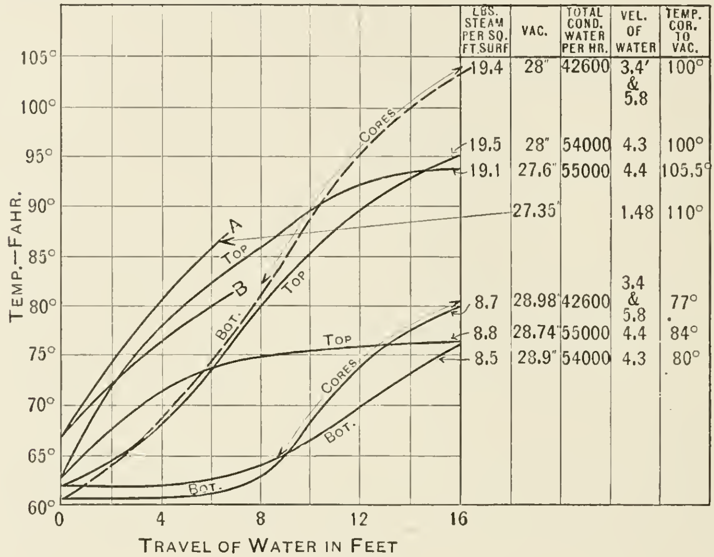


FIG. 74.

from the experiments made by Professor Weighton upon a Contraflo condenser, one group of three curves being for a rate of condensation of about 19.5 pounds of steam per square foot of cooling surface per hour, and the other group for about 8.75 pounds per hour. It has been pointed out before that condensers in ordinary use have practically pure saturated steam at the top, and at the bottom air with a small amount of water vapor. When the condensing water enters the lower tubes of the condenser we should expect that curve to be similar to *B* at the outset and similar to *A* at the end, while with water introduced at the top the curve should start out with the greater inclination and end with the lesser. It will be seen from Fig. 74 that the curves marked "top" and "bottom" have this difference in character. It will be noticed that the curves for the greater

rate of condensation are steeper than those for the lesser rate, and in some places steeper than curve A. This shows the beneficial effect of velocity of steam flow over the tubes as this prevents the tubes from being blanketed by the air which tends to collect around them. At the greater rate of condensation the velocity of the mixture in the spaces between the tubes is greater than with the lower rate of condensation.

143. Tube Length. — The curves of Fig. 74 may be used also to illustrate the effect of added tube length, or increase of surface for a given amount of steam to be condensed. Any surface added to a condenser is, in effect, added at the bottom where the air predominates, and if the original surface was sufficient the added surface simply increases the weight and cost of the condenser. If a condenser has a rate of condensation of 19.5 pounds per square foot and the surface is increased so that the rate is only about 8.75 pounds per square foot, it will be seen from Fig. 74 that about one-third of the surface is useless since there is hardly any increase in the temperature of the water passing through the bottom tubes. If the velocity of the circulating water had been decreased in the latter case the curve would not have been so flat in the region of the bottom tubes and the size of the circulating plant could have been decreased, but 4.5 feet per second is not an excessive velocity and much more would be gained by decreasing the cooling surface.

144. Rate of Heat Transmission. — Another point brought out by Fig. 74 is the low rate of heat transmission for air. The water passing through the tubes at a temperature of about 60° F. extracted but little heat from the air in the bottom of the condenser which must have been at a temperature of 100° or more. There is lack of agreement in the results of experiments which have been made upon the rate of heat transmission for pure steam but roughly speaking we may expect about 1000 B.t.u. per hour per square foot of cooling surface per degree difference of temperature Fahrenheit, when the cooling water has a velocity of 3 feet per second. The rate of heat transmission for air under the conditions which exist in a condenser would probably not be more than 0.5 B.t.u. per hour per square foot of cooling surface

per degree difference of temperature Fahrenheit, when the air has a velocity of 20 feet per second over the cooling surface.

145. Velocity of Cooling Water. — The effect of the velocity of the cooling water upon the rate of condensation is shown by Fig. 75. Curves *A* and *B* are from Mr. Smith's experiments, *A*

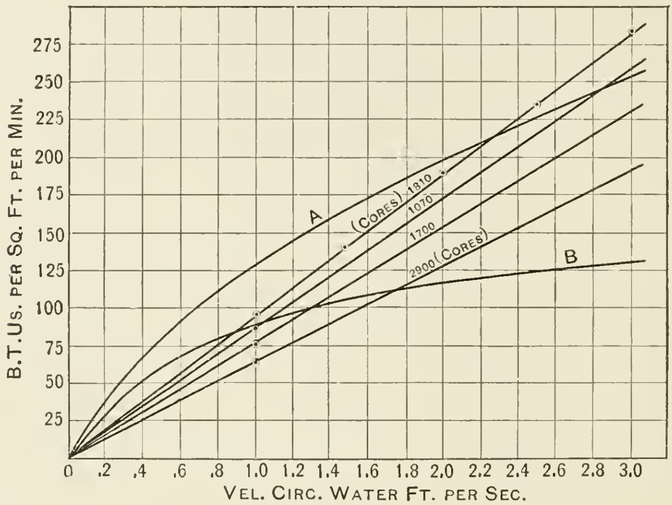


Fig. 75.

for practically pure saturated steam and *B* for a mixture of steam and air such that the air contributed 0.22 inch mercury to the total pressure, the temperature being 110° F. The initial temperature of the cooling water was 70° F. and the surface section ratio was 553. The other curves were obtained from Professor Weighton's experiments on Contraflo condensers. The fact that the curves from condenser tests are practically straight lines shows the beneficial effect of velocity of flow of gases over the cooling surfaces. In curves *A* and *B* there was practically no movement of the gases across the cooling surface and as the rate of condensation increased the air film around the tubes thickened and the rate of increase of heat transference was not proportional to the rate of increase of the velocity of the cooling water. In the Contraflo condenser the sweep of the gases over the cooling surface prevented to a large extent the formation of

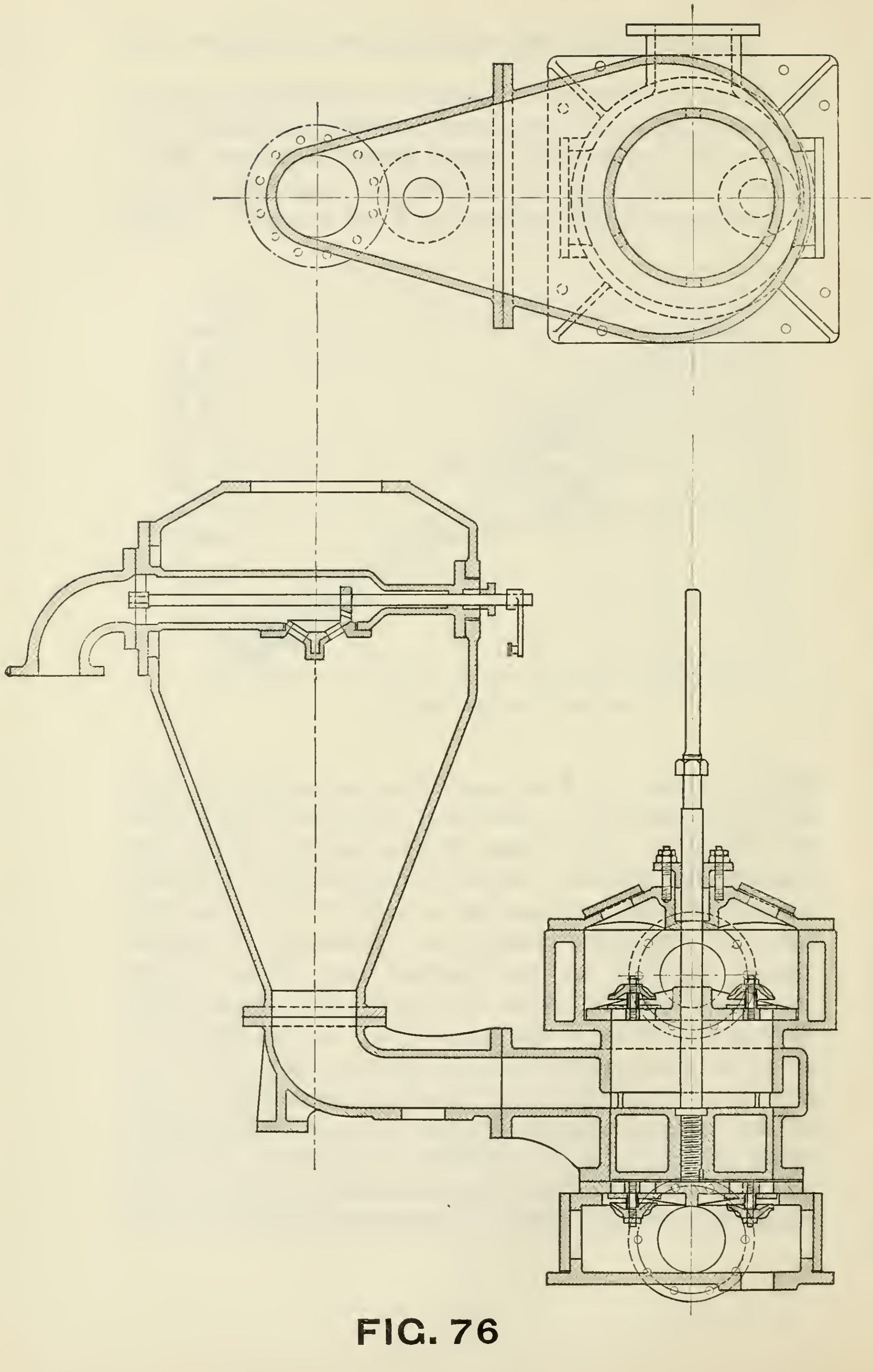
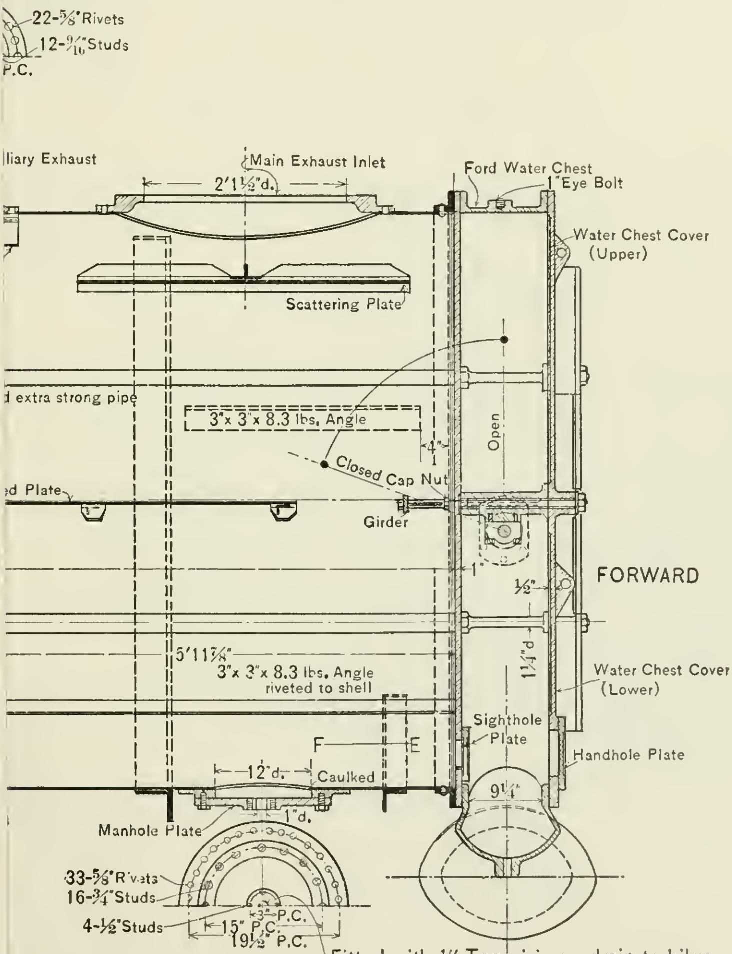


FIG. 76

NOTE:

Condenser Shell and Tube Sheets to have red lead joints.
All other joints, red lead and canvas.

4068 Tubes
7986 Sq. Ft. of Cooling Surface.



Fitted with 1" Tee giving a drain to bilge and a steam inlet for boiling out.
There will be a stop valve on each connection.

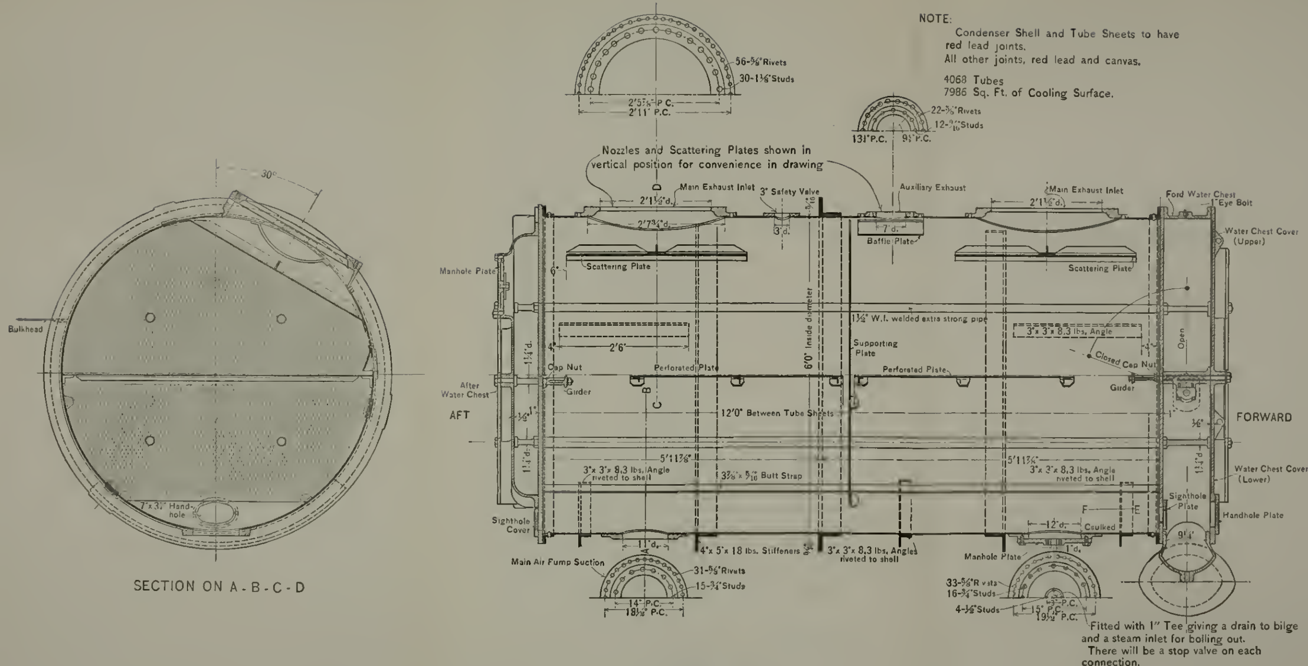

NOTE:
Condenser Shell and Tube Sheets to have red lead joints.
All other joints, red lead and canvas.
4068 Tubes
7986 Sq. Ft. of Cooling Surface.
56-5/8" Rivets
30-1 1/8" Studs
2'11" P.C.
22-5/8" Rivets
12-9/16" Studs
Nozzles and Scattering Plates shown in vertical position for convenience in drawing
Main Exhaust Inlet
3" Safety Valve
Auxiliary Exhaust
Main Exhaust Inlet
Ford Water Chest
Eye Bolt
Water Chest Cover (Upper)
Baffle Plate
Scattering Plate
Scattering Plate
Manhole Plate
Bulkhead
1 1/2" W.I. welded extra strong pipe
3"x 3"x 8.3 lbs. Angle
Supporting Plate
6'0" Inside diameter
Open
Closed Cap Nut
Cap Nut
Perforated Plate
Perforated Plate
After Water Chest
Girder
Girder
AFT
12'0" Between Tube Sheets
FORWARD
3"x 3"x 8.3 lbs. Angle riveted to shell
3 7/8"x 5/16" Butt Strap
3"x 3"x 8.3 lbs. Angle riveted to shell
Water Chest Cover (Lower)
Sighthole Plate
Handhole Plate
7"x 3 1/2" Hand-hole
Sighthole Cover
Caulked
4"x 5"x 18 lbs. Stiffeners
3"x 3"x 8.3 lbs. Angles riveted to shell
Manhole Plate
Main Air Pump Suction
31-5/8" Rivets
15-3/4" Studs
14" P.C.
18 1/8" P.C.
33-5/8" Rivets
16-3/4" Studs
4-1/2" Studs
15" P.C.
19 1/2" P.C.
Fitted with 1" Tee giving a drain to bilge and a steam inlet for boiling out.
There will be a stop valve on each connection.
SECTION ON A-B-C-D

FIG. 77

this film and the rate of condensation increased directly as the velocity of flow of the cooling water. The same curves show also the increase of rate of condensation with decrease of surface section ratio. This increased condensation is obtained, however, at the expense of circulating water, more water per pound of steam being required as the surface section ratio becomes smaller.

CONDENSERS

146. Jet Condensers. — There are two types of condensers in common use, the Surface and the Jet. In the jet condenser, see Fig. 76, the steam is condensed by mixing with a stream of water sprayed into the condenser in a finely divided form. The condensing water and the condensed steam collect at the bottom of the condenser and are pumped out by the air pump. The condenser must be large enough in the neighborhood of the water spray to enable the steam and water to become thoroughly mixed, and beyond that point the cross-sectional area can be gradually decreased until it is equal to the area of the air-pump suction pipe. There is no way of figuring the size of the condenser from theoretical considerations, but from experience we find that its volume should be about one-third of the volume of the L.P. cylinder. This type of condenser costs much less than the Surface, is smaller and can be used in vessels running in fresh water. The vacuum obtained is usually lower than that given by the surface condenser, due to the large amounts of air which come in with the condensing water. The vacuum usually obtained is around 22 inches or about 4 pounds absolute.

147. Surface Condensers. — The surface condenser, see Fig. 77, is so constructed that the steam is condensed by coming in contact with the surface of tubes through which cold water is circulated. In this way the condensed steam and condensing water are kept separate and the condenser can be used where the condensing water is not of a character suitable for feed water. This type has the further advantage that less air is introduced into the system, due to the fact that the same feed water is used over and over again and the amount of air which it contains is not as large as would be the case if a new supply were constantly

introduced. The air separates from the water in the boiler and passes through the engine and condenser into the air pump where a certain amount of it is reabsorbed by the water during compression.

About 2 per cent of make-up feed has to be added to make up for losses at leaky joints, at whistle, and in blowing off scum. This water will have more air in it than that delivered from the air pump. However, the conditions are better for a high vacuum in the surface condenser than in the jet and a vacuum of 26 inches is easily obtained. The surface condenser is more costly in construction, heavier, occupies more space, and requires more room for overhauling. It is used in vessels running in salt water and also whenever high vacua are desired.

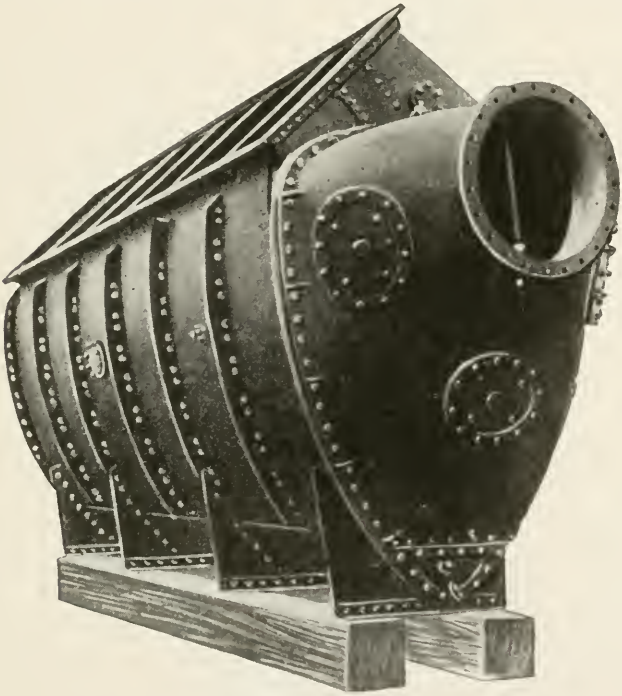
The amount of cooling surface needed per I.H.P. varies with the vacuum desired, with the construction of the condenser, and with the temperature and velocity of the cooling water. In the *old* type of surface condenser where a vacuum of 26 inches is desired it is found that 1 to 1.25 square feet of cooling surface per I.H.P. is usually sufficient in the Temperate Zone, while in the Tropics the ratio increases to 1.75 square feet per I.H.P.

148. Efficiency of Cooling Surface. — More attention has been given to condenser design lately and all designers agree that it is essential that there shall be no dead spaces in the condenser, but that the steam and air shall have as nearly as possible a uniform speed of flow over the entire cooling surface. To this end the cross-sectional area is decreased in passing from the steam inlet to the air pump suction, either by making the condenser narrower at the bottom, or by means of diaphragms so placed that the area for passage of the steam and air continually decreases.

The efficiency of the cooling surface increases with the velocity of the gases passing over it but since the increased velocity is obtained only by an increased difference in pressure between the air pump suction and the steam inlet this increased velocity means a greater back pressure in the L.P. cylinder and cannot be carried too far.

Another principle which the makers of the Contraflo con-

denser claim is of great importance is the prevention of dripping. It is claimed that if the water from the steam condensed by the top rows of tubes is allowed to drip over the lower rows these tubes serve merely to cool the feed-water and do not condense steam as it has no access to them. To obviate this diaphragms are so placed in the condenser that the water of condensation is



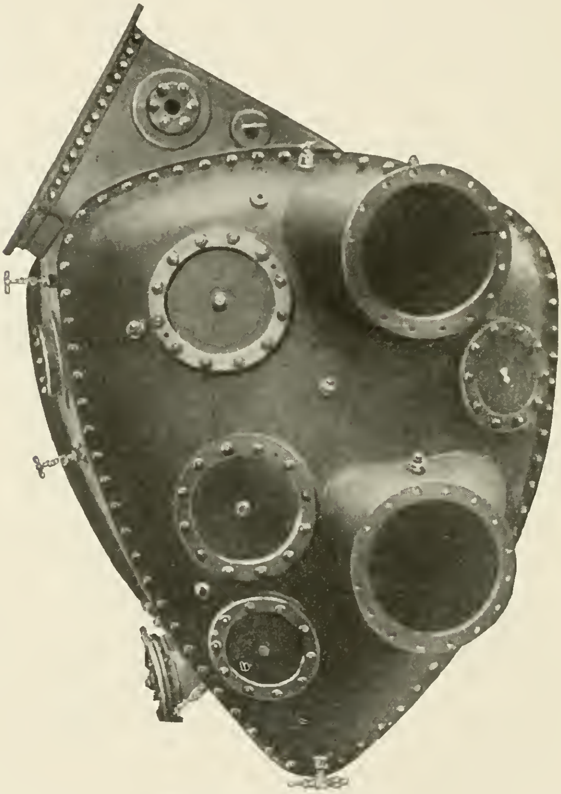
Weir-Uniflux Condenser

collected and carried off to one side and then down to the air pump suction.

The condenser cannot be considered separate from the air pump as the efficiency of the latter reacts upon the former. If the air pump is not of sufficient capacity to keep the condenser free of air the lower tubes will be drowned in air and the steam will have no access to them.

The rate of condensation will also be affected by the velocity of flow of the cooling water through the tubes. There seems to

be no reason why the rate of condensation, in a properly designed condenser, should not vary directly as the velocity of circulating water through the tubes. A greater velocity of flow means more work for the circulating pump but a slight increase in the size



Weir-Uniflux Condenser

and power of this pump may make a large saving in weight and cost of condenser.

The attention paid by modern designers to these conditions which make for a more efficient cooling surface has made it possible to reduce the size of condensers so that now the ratio of cooling surface to I.H.P. varies from 0.5 to 1, depending upon the temperature of inlet water and velocity through the tubes.

149. Comparison of Old and New Types of Condensers. — A comparison of the performances of the old and new type of condenser is made possible through the extensive experiments carried out by Professor Weighton upon a Contraflo condenser and one of the ordinary type. These experiments are published in the Transactions of the Institute of Naval Architects, Vol. 48,

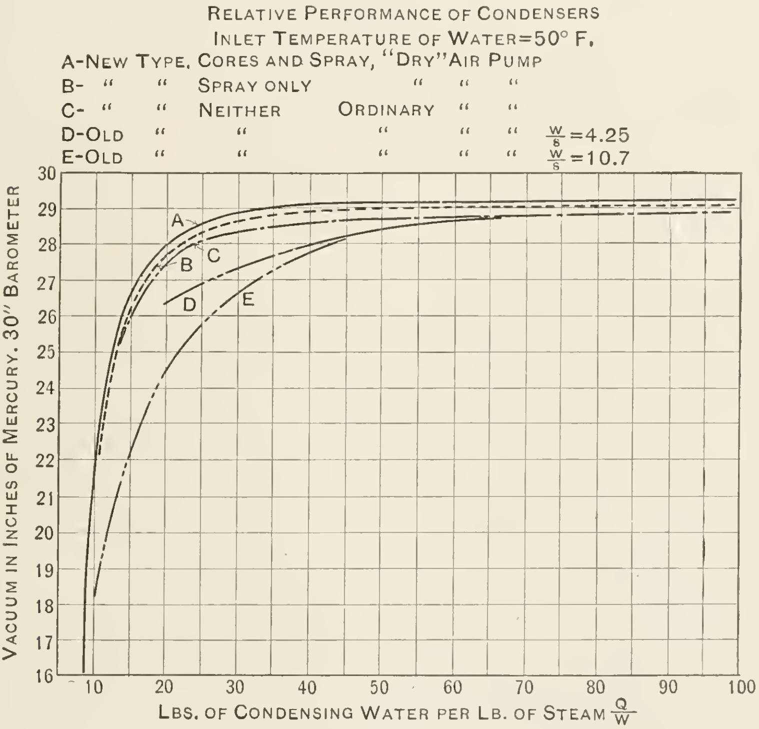


FIG. 78.

1906. Curves *C*, *D*, and *E* of Fig. 78 show the relative performance of two condensers, one of the Contraflo type and the other of the old type found in many ships, with the condenser a part of the back framing. The latter condenser had $\frac{3}{4}$-inch tubes 4 feet long, with a cooling surface of 170 square feet. The water made two passes through the condenser, giving a tube length of 8 feet. The Contraflo condenser had $\frac{5}{8}$-inch tubes, 4 feet long with 100 square feet of cooling surface. The water made four

passes through the condenser giving an effective tube length of 16 feet. With the old type it was found that there was a separate curve for each rate of condensation, the surface becoming less efficient as the rate increased. In the new type the surface efficiency was practically constant and one curve served for all rates of condensation.

150. High Vacua. — With the introduction of the steam turbine the necessity arose for a high vacuum, or low back pressure, in order that the turbine's highest efficiency might be obtained. With the reciprocating engine the increase in efficiency is considerable as the back pressure decreases up to a certain point, but with the turbine the increase is much greater. The economy in steam consumption of the reciprocating engine increases about 1.5 per cent for each inch increase in vacuum, while with the turbine an increase of vacuum from 26 to 27 inches gives a gain of about 4 per cent, from 27 to 28 inches a further gain of 5 per cent, and from 28 to 29 inches a further gain of 6 or 7 per cent in economy.

The condenser for the reciprocating engine is usually designed to give a vacuum of about 26 inches and the discharge from the air pump will be 115° or more depending upon the type of air pump used. While the efficiency of the engine alone increases as the back pressure decreases, the decrease in temperature of the water discharged by the air pump may counterbalance this gain when we consider the efficiency of engine and boiler together. At low pressures the decrease in temperature is large for a relatively small decrease in pressure. The condenser for the turbine is designed to give the highest practical vacuum, which is about 28.5 inches of mercury with sea water of 60° F. In the case of the turbine the increase in economy due to decreased back pressure is sufficiently great to more than overcome the thermal loss due to decreased temperature of the air pump discharge. With this vacuum and the large area for the passage of exhaust steam from turbine to condenser the steam can be expanded down to about 1 pound absolute.

151. Means Employed to Obtain High Vacua. — The means employed to obtain low condenser pressures may be considered

under four heads: (1) the wet and dry air-pump system, (2) an augmentor condenser, (3) two-stage air pumps, and (4) rotary air pumps. In the wet and dry air-pump system a wet pump is employed to remove the hot water and a so-called "dry" air pump is used to remove the air. In this way the condensed steam can be taken off at as high a temperature as possible and used for feed-water, while the "dry" air pump is supplied with a small amount of cold water which cools the air, seals the valves, and fills the clearance spaces. It is impossible to create a vacuum in the barrel of an air pump in the presence of water whose temperature is very close to the temperature corresponding to the vacuum desired. The evaporation of this hot water will be so rapid, as the pressure is decreased in the barrel, that the vacuum will be spoiled. Let us take the case of a turbine expanding down to 1 pound absolute. The average pressure in the condenser will be about 0.75 pound absolute and the air pump will have to develop a vacuum of about 0.5 pound absolute in order to cause the air to flow into the pump. The temperatures corresponding to these pressures are 102°, 92°, and 80°, respectively. The largest part of the steam will be condensed at the higher temperature, about 100°, and unless the water is cooled down to about 75° it will be difficult to obtain a pressure of 0.5 pound absolute in the air pump. In a condenser of the ordinary type developing a vacuum of 25 inches, or about 2.5 pounds absolute, with a single air pump, the temperature of the discharge water will be about 115°, although the temperature corresponding to 2 pounds absolute, the pressure in the pump barrel, is about 125°; showing that the temperature of the water in the air-pump barrel is some 10° less than that corresponding to the vacuum.

In some of the newer types of condenser this separation is effected by having two nozzles, one for the condensed steam and one for the air. The lower tubes of the condenser are used for cooling the water supplied to the dry air pump. Fig. 78 shows the effect of the wet and dry air-pump system upon the efficiency. Curve *B* is for the condenser with a wet pump to remove the hot water and a spray of cold water introduced into the air suction

pipe of the dry air pump. Curve C is for the same condenser with a single air pump to remove the air and water together. The water used in the spray is included in the pounds of water per pound of steam.

152. Augmentor Condenser. — Condensers to be used with Parsons turbines are often fitted with an augmentor as shown in Fig. 79. At A there is an ejector which picks up the air in the

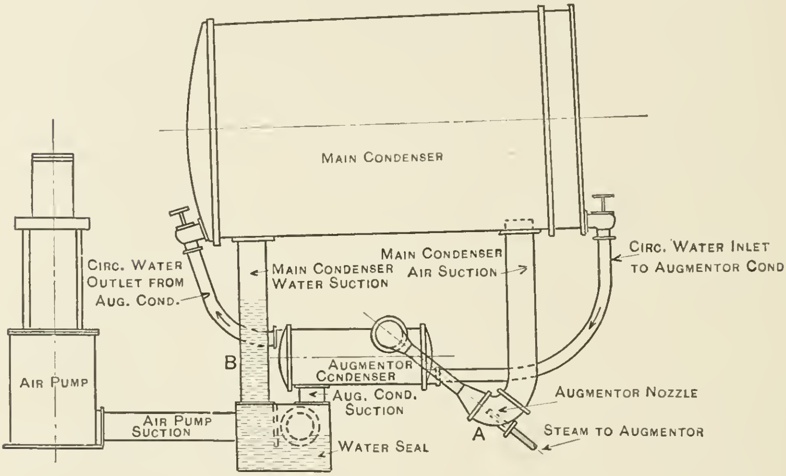


Fig. 79.

condenser at low pressure and delivers it to the air pump at a higher pressure, the difference in pressure between the air pump and condenser being maintained by the column of water B on one side and by the momentum of the moving stream of air and steam on the other. In this way an air pump which is capable of maintaining a vacuum of only 26 or 27 inches in the air-pump barrel, can by the assistance of the ejector handle a sufficient quantity of air to maintain a vacuum of 28 inches or 28.5 inches in the condenser. According to Mr. Gerald Stoney, the use of the augmentor increases the economy 7 per cent while the steam used is only 7 to 8 per cent of the steam saved.

153. Two-stage Air Pumps. — In the two-stage air pumps the compression from the low vacuum pressure up to the atmospheric pressure is performed in two or more stages. The Weir

"Dual" Air Pump is an example of this. These pumps and the rotary air pump will be considered later under the head of Air Pumps.

154. Neilson's Formula for Condenser Design. — The amount of cooling surface which a condenser should have in order that it may produce the desired vacuum should be determined by the steam consumption of the engine, the back pressure in the L.P. cylinder, the temperature of the cooling water at inlet, the velocity of the cooling water through the tubes, and the pounds of steam that can be condensed per square foot of cooling surface per hour per degree difference of temperature of cooling water and steam. There are hardly any published tests of ordinary condensers in which all these quantities were observed. Mr. Neilson read a paper before the Institute of Engineers and Ship-builders in Scotland, in February, 1910, entitled "The Design of Surface Condensers," in which he gave the following equation:

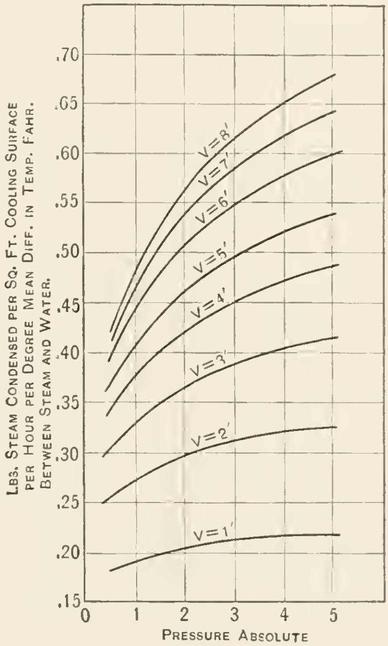


Fig. 80.

$$W = Sk\theta_m. \tag{72}$$

W = weight of steam condensed per hour.

S = square feet cooling surface in tubes.

K = pounds of steam condensed per hour per square foot of surface per degree mean difference of temperature of steam and condensing water.

$$\theta_m = t_v - \frac{t_i + t_o}{2} - C.$$

t_v = temperature ° F. corresponding to vacuum desired.

t_i = temperature ° F. of condensing water at inlet.
t_o = temperature ° F. of condensing water at outlet.
C = 2.5 for ordinary condensers.

Mr. Neilson gives a formula for calculating K and the curves of Fig. 80 have been drawn by means of this formula. In the same paper the quantity of cooling water necessary per hour is given as

$$Q = \frac{1000\, W}{t_o - t_i}. \tag{73}$$

The values of t_0 to be used in these equations are given by the curve of Fig. 81.

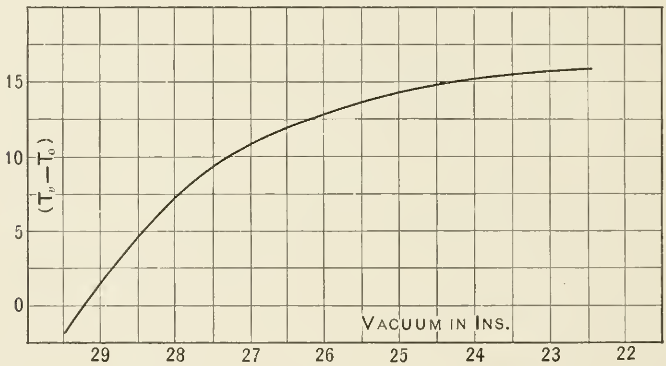


FIG. 81.

155. Weighton's Experiments. — A very complete set of experiments upon the newer type of condenser was carried out by Professor Weighton and the results are published in "The Transactions of the Institute of Naval Architects," Vol. 48, 1906. The results there given have been put into a different form by the author and are shown in Fig. 82. The experiments were made with condensing water whose inlet temperature varied from 45° to 70°. Variation was also made in the surface-section ratio, or ratio of the surface of the tube element exposed to steam to the cross-sectional area of the water passage through the tube. Two of the experiments were upon condensers whose surface-section ratios were 1070 and 1700, run in connection with a single

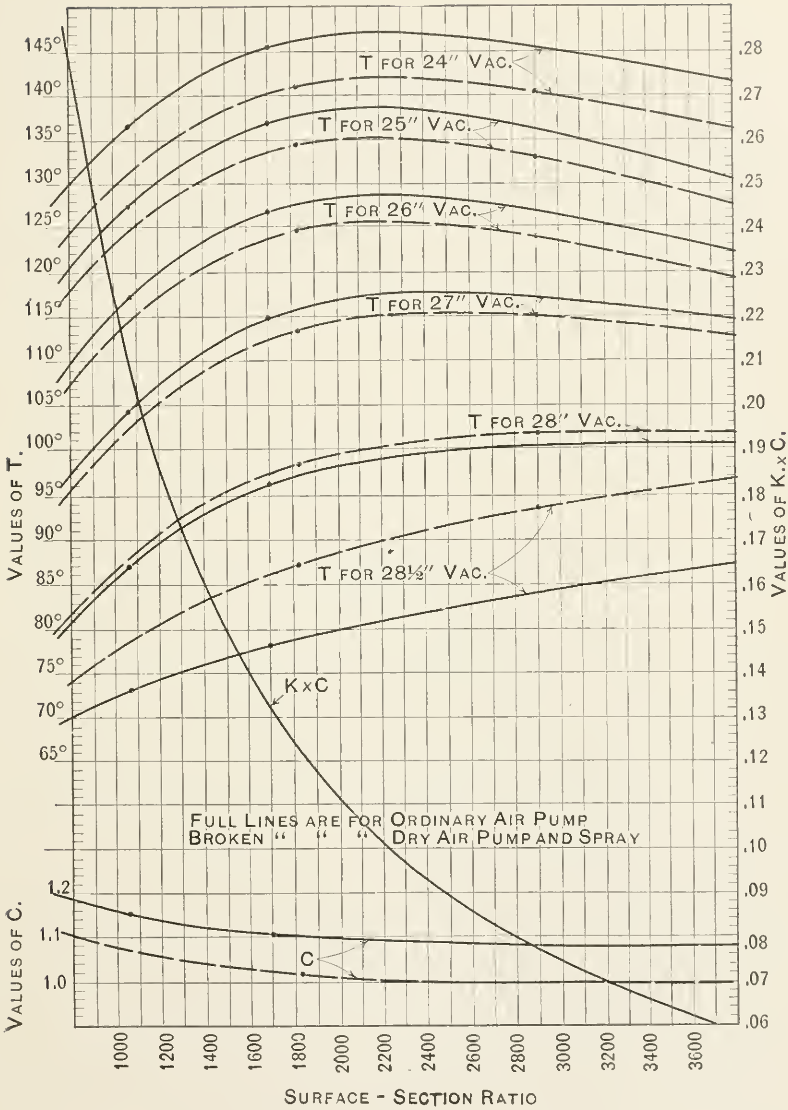

T FOR 24″ VAC.
T FOR 25″ VAC.
T FOR 26″ VAC.
T FOR 27″ VAC.
T FOR 28″ VAC.
T FOR 28½″ VAC.
K×C
C
FULL LINES ARE FOR ORDINARY AIR PUMP
BROKEN " " " DRY AIR PUMP AND SPRAY
VALUES OF T.
145° 140° 135° 130° 125° 120° 115° 110° 105° 100° 95° 90° 85° 80° 75° 70° 65°
VALUES OF C.
1.2 1.1 1.0
VALUES OF K.×C.
.28 .27 .26 .25 .24 .23 .22 .21 .20 .19 .18 .17 .16 .15 .14 .13 .12 .11 .10 .09 .08 .07 .06
1000 1200 1400 1600 1800 2000 2200 2400 2600 2800 3000 3200 3400 3600
SURFACE - SECTION RATIO

FIG. 82.

air pump of the ordinary type. Two other experiments were upon the same condensers with triangular wooden cores placed in the tubes so that the surface-section ratios were 1810 and 2900. In these experiments the condenser was run in connection with a wet and a dry air pump, and a spray of cold water was introduced into the air-pump suction.

156. A Method of Design Based upon Weighton's Experiments. — The curves of T as drawn in Fig. 82 are determined by only two points but it was assumed that the difference in air pumps would cause only a bodily displacement of one curve from another and would not cause the curves to differ materially in character, so that the points for one could be used as a guide in fairing in the other. The curves were derived as follows:

Let
$$W = KSv^x(T - T_i), \tag{74}$$
and
$$Q = C\frac{1000\,W}{T - T_i}. \tag{75}$$

W_1 = pounds of steam condensed per hour.

W = equivalent pounds of steam at 8 pounds absolute and hot-well temperature of 100° F.

$$= \frac{\text{total heat at release minus heat of hot-well}}{1070}W_1.$$

K = factor to be determined from experiments.

S = cooling surface of tubes in square feet.

v = velocity of condensing water through tubes in feet per second.

x = exponent of v, to be derived from experiment.

T = factor to be derived from experiment.

T_i = temperature ° F. of condensing water at inlet.

Q = quantity of condensing water per hour in pounds.

C = factor to be derived from experiment.

From (74) and (75) we can derive the following:

$$K = \frac{W}{S}\frac{1}{v^x}\frac{1}{T - T_i}, \text{ and } C = \frac{Q}{W}\frac{T - T_i}{1000}.$$

$$KC = \frac{W}{S}\frac{1}{v^x}\frac{1}{T - T_i}\frac{Q}{W}\frac{T - T_i}{1000} = \frac{Q}{S}\frac{1}{v^x}\frac{1}{1000}. \tag{76}$$

The velocity of the water in the tubes will be

$$v = \frac{Q}{62.15 \times 3600 \times A}.$$

62.15 = mean between the weight of a cubic foot of water at 50° F. and 110° F.

A = total area in square feet of all the tube elements for the passage of water. The number of tube elements is equal to the total number of tubes divided by the number of times the water passes through the condenser.

$$A = \frac{S}{s}.$$

s = surface section ratio = area in square feet of the cooling surface of one tube element divided by the area in square feet of the water passage in tube element.

$$v = \frac{Qs}{223{,}740\, S}.$$

$$\frac{Q}{S} = \frac{223{,}740\, v}{s}. \tag{77}$$

$$\therefore \quad \frac{Q}{W} \cdot \frac{W}{S} = \frac{223{,}740\, v}{s}. \tag{78}$$

Formula (78) contains some of the quantities usually observed in experiments and gives the relation that these observed quantities should bear to one another. $\frac{Q}{W}$ = the quantity of water per pound of steam, and $\frac{W}{S}$ = the pounds of steam condensed per square foot of cooling surface.

From (76) and (77) we get

$$KC = \frac{223{,}740\, v}{s} \frac{1}{v^x} \frac{1}{1000} = v^{1-x} \frac{223.74}{s}.$$

If $x = 1$,

$$KC = \frac{223.74}{s}. \tag{79}$$

If the tubes of a condenser are clean, if it is so designed that there are no dead spaces, and if the air pump is of sufficient

capacity to keep the lower tubes from being drowned in air, the value of x should be 1; i.e., the rate of condensation should vary directly as the velocity of the condensing water in the tubes. It is not impossible for x to be greater than 1 and in some tests of the Contraflo condensers this was true. The more rapid the circulation of water the less likelihood is there of a solid core of water going through the tube without coming in contact with the sides.

The curves of Fig. 80 calculated from the formula given by Mr. Neilson for the old type of condenser show that the value of x is about 0.42 for a 29-inch vacuum, about 0.5 for a 26-inch vacuum, and about 0.55 for a 22-inch vacuum. This formula contains a factor which makes allowance for the tubes being dirty. It would appear, then, that under the most favorable conditions the value of $x = 1$, and under the most unfavorable $= 0.5$.

Since these relations exist between the different factors the condensers can be designed by means of the curves of Fig. 82 and the following formulæ:

$$Q = C \frac{W\,1000}{T - T_i}. \tag{75}$$

$$S = \frac{Qs}{223{,}740\,v}. \tag{77}$$

$$A = \frac{S}{s}. \tag{80}$$

$$\frac{A}{a} = \text{number of tube elements.} \tag{81}$$

a = area in square feet of the water passage in one tube element.

The length of tubes will depend upon the number of times the water is to pass through the condenser.

According to Formula (79) the product KC is constant for $v = 1$ and is equal to $\frac{223.74}{s}$ irrespective of the value of x. It should not be inferred from this that K and C are constant. If the tubes are dirty so that $x < 1$, more water will be required to

condense a given amount of steam and the value of C will be larger and K will be smaller. Allowance for dirty tubes or ineffective surface can be made by increasing the values of C derived from experiments on condensers with clean tubes.

The curves of Fig. 82 are primarily for the type of condenser experimented on and do not necessarily apply to any other type of condenser.

The values of T are so chosen that for any given vacuum the values of K, or C, determined from the results of the test, very nearly coincide for the different values of T_i. For instance, take the following case:

No. 2 condenser, vacuum = 27 inches, v = 2.

T is arbitrarily chosen to be 114°.

T_i	$\frac{W}{S}$	$\frac{W}{S(T - T_i)}$	K
45	15.5	0.225	0.113
55	13.5	0.229	0.115
65	11	0.224	0.112
70	10	0.227	0.114

The values of K and C given by the curves of Fig. 82 do not in the worst case differ more than 2 per cent from the values given by Professor Weighton, and in most cases are closer than 2 per cent.

The values of K decrease as the surface-section ratio increases so that a larger amount of surface must be used for the larger surface-section ratios. The quantity of water necessary to condense 1 pound of steam will decrease as the surface-section ratio increases. Where water is cheap it is better to use a low surface-section ratio, say in the neighborhood of 2000, as by so doing the weight of the condenser can be cut down considerably while the extra power required of the circulating engine will not increase its weight materially. Where water is expensive a higher surface-section ratio of 3000 should be used.

157. Velocity of Cooling Water. — The weight of the condenser can be reduced by running the water through the tubes at a higher velocity. It is probable in the old types of condenser

this velocity has been as low as 1 foot per second under ordinary conditions. A velocity of 3 or 4 feet per second can be used ordinarily and still not throw an undue load upon the circulating engine if the velocity has to be increased 1 or 2 feet per second when the tubes get dirty or the inlet temperature high.

The condensing water tends to pass through the tubes with a central core more or less unaffected by the walls of the tubes. As the quantity of condensing water is increased the velocity of

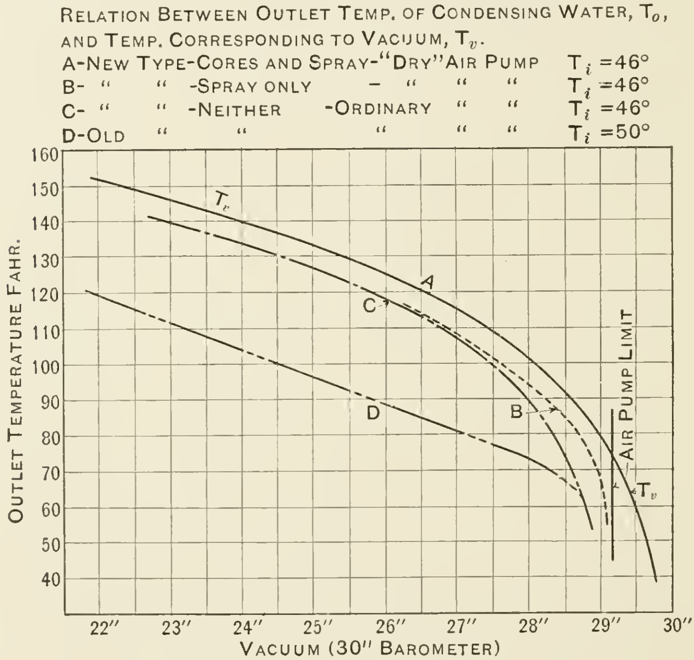


Fig. 83.

the central core increases faster than the velocity of the water in contact with the tube walls. This will cause more or less eddying and cross-currents due to pressure differences and will bring the water at the center in contact with the tube walls to a certain extent. The greater the length of the tubes the greater will be the resistance to flow along the tube walls and the greater will be the eddying and cross-current effect. This central core of unaffected water can also be prevented by placing a tri-

angular or other shaped core in the tube, as in this way a comparatively thin layer of water passes through. Curve *A* of Fig. 83 shows the increase in efficiency due to this device. These cores increase the work of the circulating pump and care must be taken that the velocity chosen is not so high that this increased work neutralizes the gain from the increased vacuum. This device could be used where condensing water was scarce and free from sediment and the extra power demanded of the circulating pump would cost less than the water saved. The effect of these various conditions upon the outlet temperature of the condensing water is shown by Fig. 83.

With cores in the tubes and the dry air pump cooled by the water spray the final temperature was the same as that due to

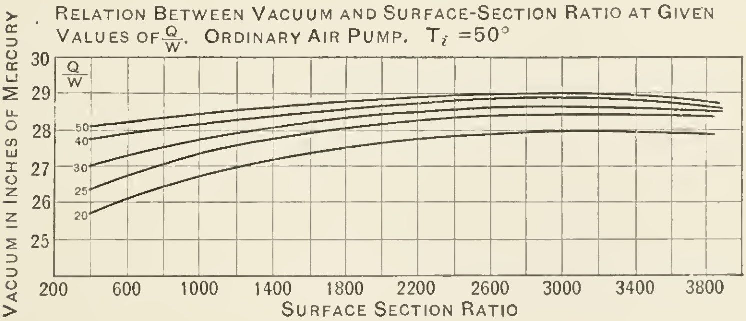


Fig. 84.

the vacuum at the condenser top. This does not mean that the condensing water and steam are at the same temperature for it is found that the temperature at the top of the condenser is always in excess of that due to the vacuum registered. The low value of T_0 for the old type of condenser was due to the large diameter of the tubes relative to their short length.

158. Effect of Surface-section Ratio. — In the experiments of Professor Weighton it was found that the surface-section ratio affected the efficiency in the manner shown by the curves of Fig. 84. The maximum efficiency seems to be obtained with a surface-section ratio of about 3000. According to these curves a vacuum of 28 inches could be obtained with 20 pounds of

water per pound of steam and a surface-section ratio of 3000; or with 25 pounds of water per pound of steam at a surface-section ratio of 1700; or with 30 pounds of water per pound of steam and a surface-section ratio of 1300, etc.

A larger surface-section ratio would be accompanied by a greater weight and first cost of condenser, and greater first cost of circulating engine, but by increased economy in condensing water. If water is cheap it would be better to use a smaller surface-section ratio than that giving maximum efficiency in order to save in weight and first cost of condenser and circulating engine.

159. Effect of Admitting Water at Top and Bottom of Condenser. — It is common practice on shipboard to admit the

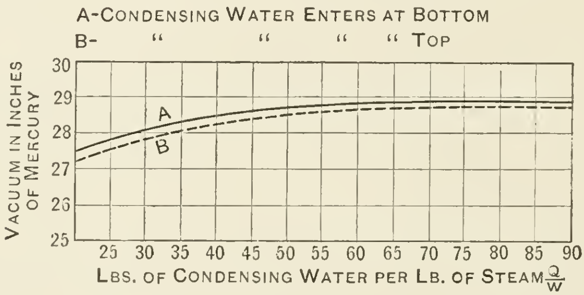


Fig. 85.

condensing water to the top tubes of the condenser and discharge it from the bottom tubes in order that the water from the condensed steam may not be chilled down by the cold entering water. It was found by test, however, to be more economical to have the water enter the lower tubes and discharge from the upper tubes. Curves *A* and *B* of Fig. 85 show the relative effect of the two methods of circulation. The steam entered the top of the condenser in these experiments.

160. Admission of Steam to Condenser. — Condensers are sometimes made with the steam entering the bottom and the air is taken out at the top. The steam condensed on the lower tubes drips down and meets the incoming exhaust and is heated by it so that the temperature of the discharge to the feed tank is

quite high. This method makes the condenser a partial jet condenser. The Alberger Condenser Company make a condenser of this type, the hot water being taken out of the bottom of an elbow in the exhaust pipe.

Condensers have been made with the steam passing through tubes surrounded by water but this method puts too high a back pressure on the engine, as the resistance to the flow of steam is large, and it is not easy to allow for the constant decrease in volume occupied by the steam and air. This causes the velocity to be large at entrance and small at exit, whereas a uniform velocity is desirable.

161. Sizes of Condenser Tubes. — Condenser tubes are usually $\frac{5}{8}$ or $\frac{3}{4}$ inch in diameter and the thickness is 18 to 20 B.W.G., or 0.048 to 0.036 inch. When $\frac{5}{8}$-inch tubes are spaced $\frac{15}{16}$ inch apart the number of tubes per square foot of tube plate is about 135; when spaced 1 inch apart, about 125. This includes space for shell clearance, steam passageways, etc.

A paper by Mr. William Weir in the "Transactions of the Institution of Engineers and Shipbuilders in Scotland," Oct. 22, 1912, gives the following information concerning condenser weights.

	Vacuum, inches	Sea temperature, °F.	Shell material	Type of condenser	Weight per I.H.P., pounds	Type of engine
Cargo steamer 3000 H.P.	23	85	C.I.	In engine frame	8.3	Recip.
Cargo steamer 3000 H.P.	23	85	Steel	Circular	7.5	Recip.
Cargo steamer 3000 H.P.	25	85	Steel	Uniflux	3.4	Recip.
Cross channel	28.5	55	Steel	Uniflux	3.5	Turbine
Atlantic liner	28.5	60	Steel	Uniflux	4.7	Turbine
Destroyer (1900)	26.5	55	Brass	Oval	2.7	Turbine
Destroyer (1912)	28	55	Steel	Uniflux	2.1	Turbine
Cruiser (1907)	27	55	Gun metal	Circular	4.4	Turbine
Cruiser (1912)	28.3	55	Steel	Uniflux	3.4	Turbine
Battleship (1906)	28.5	55	Gun metal	Circular	6.1	Turbine
Battleship (1912)	28.5	55	Steel	Uniflux	4.1	Turbine

AIR PUMPS

162. Relation of Air Pump and Condenser. — The air pump and the condenser are so intimately related to one another that they should be considered as a unit rather than as two separate auxiliaries. This is illustrated by Mr. Neilson's statement:

"The function of the condenser is to reduce the ratio of steam to air by the condensation of the greater part of the steam, so as to diminish the volume of mixture of air and steam which has to be discharged by the air pump. The function of the air pump is to keep down the ratio of air to steam in the condenser, so as to allow the cooling surface of the latter to act effectively."

163. Neilson's Diagram. — We have already considered the effect of air in the condenser upon the rate of condensation. The effect of vapor in the mixture delivered to the air pump is shown by a diagram constructed by Mr. Neilson, see Fig. 86. It will be seen that for any given vacuum as the ratio of air to vapor decreases the volume of air to be handled by the air pump increases very rapidly. The diagram also shows the marked decrease in the volume to be handled by the air pump caused by lowering the temperature for a given vacuum. This again emphasizes the desirability of intercepting the condensed water as early as possible and removing it while hot, and then cooling the air and vapor by some means.

164. McBride's Diagram.—The same diagram can be used for the determination of the amount of air present in the steam when entering the condenser, a point discussed by Mr. McBride in a paper before the American Society of Mechanical Engineers, June, 1908. Mr. McBride assumes that one part of air to 10,000 parts of steam is present in the steam as it enters the condenser. The air bears such a small ratio to the steam that the latter will be practically saturated, and both air and steam can be considered as under the same pressure and at the temperature corresponding to that pressure. As the steam and air pass through the condenser the steam is condensed, the temperature falls, and in the air-pump suction pipe we have saturated air whose volume will depend upon the temperature, assuming the pressure to be unchanged in passing through the condenser. While at the entrance to the condenser the air would have a volume determined by the absolute pressure at that point, in the air-pump suction the air will have a volume determined by its partial pressure, and this again will depend upon the temperature of the mixture. Mr. McBride expresses this volume in terms of the

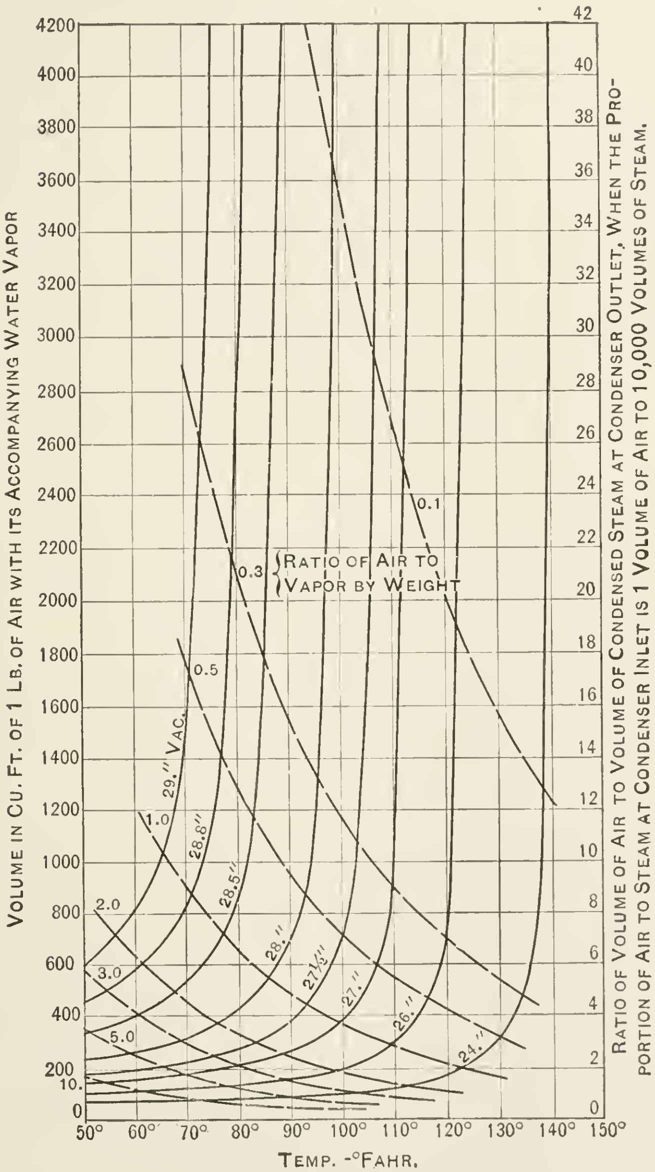

Volume in Cu. Ft. of 1 Lb. of Air with its Accompanying Water Vapor
4200
4000
3800
3600
3400
3200
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
10.
0
Ratio of Volume of Air to Volume of Condensed Steam at Condenser Outlet, When the Proportion of Air to Steam at Condenser Inlet is 1 Volume of Air to 10,000 Volumes of Steam.
42
40
38
36
34
32
30
28
26
24
22
20
18
16
14
12
10
8
6
4
2
0
Ratio of Air to Vapor by Weight
0.1
0.3
0.5
1.0
2.0
3.0
5.0
29." Vac.
28.8"
28.5"
28."
27½"
27."
26."
24."
50°
60°
70°
80°
90°
100°
110°
120°
130°
140°
150°
Temp. -°Fahr.

Fig. 86.

volume of 1 pound of condensed steam. Thus, if the vacuum at entrance to the condenser is 28.5 inches the temperature will be 92° F. and the absolute pressure 0.7368 pound. The volume of 1 pound of steam at this pressure will be about 445 cubic feet. If 1 volume of air to 10,000 volumes of steam is present the volume of air at entrance will be 0.0445 cubic foot in every pound of steam. If the pressure in the air-pump suction pipe is 28.5 inches and the temperature is 80° F., the steam will exert a pressure of 0.505 pound. The pressure of the air will then be 0.7368 − 0.505 = 0.2318 pound per square inch. The volume of the air will be

$$0.0445 \times \frac{0.7368}{0.2318} \times \frac{459.5 + 80}{459.5 + 92} = 0.138 \text{ cubic foot.}$$

The volume of 1 pound of water at this temperature will be

$$\frac{1}{62.25} = 0.01605. \quad 0.138 \div 0.0165 = 8.36.$$

The air pump would need a capacity of 9.36 × volume of steam condensed.

Mr. Neilson's diagram can be calculated from a formula derived as follows:

$$A = \frac{12.387 \times t \times 14.7}{491.5\,(P - p)} = \frac{0.3705\,t}{P - p}. \qquad (82)$$

12.387 = volume of 1 pound of air at 32° F.
P = absolute pressure of mixture in air-pump suction.
t = absolute temperature, F, of mixture in air-pump suction.
p = absolute pressure of saturated steam at temperature t.

Mr. McBride's diagram can be calculated from a formula derived as follows:

$$B = \frac{V}{10{,}000} \times \frac{t}{T} \times \frac{P}{(P - p)} \times \frac{1}{w}. \qquad (83)$$

P, t, and p are as above.
V = volume of 1 pound of saturated steam at pressure P.
T = absolute temperature of saturated steam at pressure P.
w = volume of 1 pound of water at temperature t (cubic feet).

At 50° F. $\frac{PV}{T} = 0.595, \quad \frac{1}{w} = 62.4.$

At 120° F. $\frac{PV}{T} = 0.592, \quad \frac{1}{w} = 61.7.$

Therefore at 50° F.

$$B = \frac{0.003715\, t}{P - p},$$

and at 120° F.

$$B = \frac{0.00366\, t}{P - p}.$$

For all practical purposes $B = \frac{A}{100}$.

165. Determination of Air Leakage. — We can, therefore, use Fig. 86 for the determination of air leakage into a condenser by simply dividing the ordinates by 100. For the purpose of illustration take a case where the following quantities are known: 28-inch vacuum, or $P = 0.9823$ pound absolute; $t = 95°$ F. $= 554.5°$ absolute; also the volume of water pumped; the stroke, diameter, and number of double strokes of the air pump. Allowing for slip, the volume of air and water pumped can be found and divided by the volume of water pumped; let this equal 40. Then the ratio of air to water will be 39 to 1. From Fig. 86 we see that for the vacuum and temperature given one part of air to 10,000 parts of steam would give a ratio of 12 to 1 in the air-pump suction pipe. Therefore in this case the proportion of air to steam at entrance to the condenser is 3.25 in 10,000. In this way one can determine whether the air leakage into a condenser is normal or abnormal.

166. Air Leakage Allowed for by Manufacturers. — Mr. McBride states that manufacturers of vertical twin air pumps for surface condensers for a vacuum of 26 inches, and a temperature of 110° F. in the hotwell, furnish a pump capable of displacing about 13 times the water to be pumped. From Fig. 86 we see that this allows for about 4 parts of air to 10,000 of steam. The volume would be made up about as follows:

Condensed steam..........................	1.0 volume
Air in feed-water ($1\frac{1}{2}$ per cent by volume).	0.3 volume
Air leakage	11.7 volumes
	13.0 volumes

In the case of horizontal air pumps for a 26-inch vacuum and 110° F. temperature of hotwell, Mr. McBride states that allowance is made for a displacement of about 20 times the volume of the water of condensation.

Where jet condensers are used for a vacuum of 26 inches, hotwell temperature of 110° F., injection temperature of 70° F., and discharge temperature of 110° F., the displacement of the pump is made 52 times the volume of the water of condensation. The volumes would be about as follows:

Condensed steam.......................	1.0 volume
Air in feed-water ($1\frac{1}{2}$ per cent by volume)..	0.3 volume
Condensing water	26.0 volumes
Air in condensing water (2 per cent by volume)	11.0 volumes
Air entering by leakage..................	13.7 volumes
	52.0 volumes

Mr. McBride assumes from these allowances by manufacturers that in ordinary land practice the leakage of air into engines is about in the ratio of 4.5 parts per 10,000 parts of steam.

167. Air Leakage in Delaware's Engines. — The tests of the U.S.S. *Delaware* gave data from which the following air leakages have been computed.

	Vacuum	Hot-well temperature	Pressure in 2nd rec., absolute	I.H.P.	Volumes of air per 10,000 volumes of steam	Double strokes of air pump per minute	Remarks
21.56 knots.	26.3	104.5	37	28,578	2.65	20.2	....
19 knots....	27.2	89.7	43.5	16,602	5.53	21	M.P. cut-off shortened
12 knots....	28	90.6	14.5	3,905	8.7	21.2	....

Independent air pumps are usually designed to give the desired displacement at 20 to 30 double strokes per minute.

168. Air-pump Capacity. — It is obvious that no fixed rule can be given for the size of the air pump as it will depend upon the efficiency of the condenser in removing the vapor from the air and upon the air leakage of the engine. The most fruitful sources of air are the L.P. cylinder and valve chest and the auxiliaries, if the latter exhaust below the pressure of the atmosphere. The air coming from the auxiliaries can be eliminated by putting a relief valve between the auxiliary exhaust line and the condenser which will keep the exhaust pressure above that of the atmosphere. The leakage in the L.P. cylinder is probably at the stuffing boxes of the piston rod and valve stems. It is noticed that when the engine is run at reduced power, with the admission pressure to the L.P. below atmospheric pressure, it is very difficult to keep a good vacuum. This leakage at the piston rod and valve stem packings can be eliminated by supplying the stuffing boxes with steam from the intermediate receiver so that there will be a tendency for steam to escape from the boxes rather than for air to leak in.

The proportions recommended for the air pump vary considerably as the following will show. Mr. LeBlanc says that for turbines with a 29-inch vacuum the air pump must handle 21 times the volume of water, and for a reciprocating engine with a 26-inch vacuum the pump must handle 12 times the volume of water.

McBride gives the following as stated above:

Vertical twin air pump, surface condenser, 26-inch vacuum, 110° F. hotwell. Pump must handle 13 times the volume of water.

Horizontal air pump, same condition as above. Pump must handle 20 times volume of water.

Jet condensers, same condition as above and 70° F. inlet temperature of injection. Pump must handle 52 times volume of feed-water.

Neilson gives the following:

Proportions of condenser and air pump should be such that,

$$\frac{\text{cu. ft. (work. stroke) air pump displacement per hour}}{\text{sq. ft. condensing surface}} = 4\text{–}7\tfrac{1}{2},$$

or 4 to $7\frac{1}{2}$ cubic feet air-pump displacement for 1 square foot of cooling surface. If each square foot of cooling surface condenses 20 pounds of steam this would result in allowing an air-pump displacement of 12.5 to 23 times the volume of water.

Professor Weighton's experiments showed that a vacuum of 29 inches could be produced if there was a displacement of 0.7 cubic foot for every pound of steam. This is equivalent to a displacement of 43.5 times the volume of water.

A $28\frac{1}{2}$-inch vacuum was produced with 0.3 cubic foot of air-pump displacement per pound of steam, or a displacement of 18.6 times the volume of water to be pumped.

Authority	Description	Vacuum, inches	Feed temperature	Ratio of pump displacement to water pumped
McBride....	Vert. twin pump, surf. cond.	26	110	13
McBride....	Hor. air pump, surf. cond.	26	110	20
McBride....	Jet cond. inlet water at 70°	26	110	52
Neilson....	Surface condenser, 20 pounds of steam per square foot of cooling surface	26 to $28\frac{1}{2}$	......	12.5 to 23
Weighton...	Surface condenser...........	29	......	45
Weighton...	Surface condenser...........	$28\frac{1}{2}$	......	20

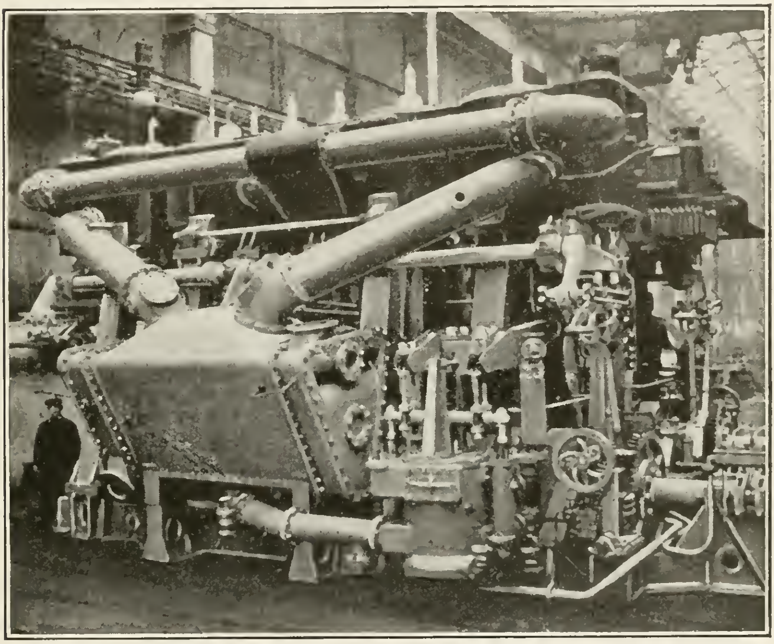
169. Attached Air Pumps. — In marine engine practice it is customary to determine the size of an attached air pump (i.e., one which is run by means of levers from one of the crossheads) by the size of the L.P. cylinder.

$$\text{Volume of air pump} = \frac{\text{volume of L.P. cylinder}}{C}. \qquad (84)$$

Triple engine, surface condenser, $C = 15$ to 20.
Triple engine, jet condenser, $C = 12.5$.
Compound engine, surface condenser, $C = 10$.
Compound engine, jet condenser, $C = 8$.

The attached air pump will always be larger than necessary for full power if it is made large enough for reduced powers.

With independent pumps it is found that the number of double strokes per minute will remain about the same for reduced powers as for maximum power. The greater air leakage which occurs at reduced powers requires a greater displacement relative to the amount of water used. If the pump is attached to the main engine its capacity will decrease as the revolutions decrease, so



Four-cylinder Triple-expansion Engine with Attached Pumps and Weir-Uniflux Condenser.

that unless it is made larger than necessary for full power it will not be sufficient at reduced powers.

170. Air-pump Proportions. — The speed of the bucket should be from 200 to 300 feet per minute, the lower speed to be used with jet condensers. The speed of the water and air through the valves will be greater than this, as the net area through the valves is usually from 22 to 25 per cent of the area of the bucket. The lift of the valves should be $\frac{1}{8}$ of the diameter. The construction of the valves is shown in Fig. 87 and their arrangement in Fig. 88.

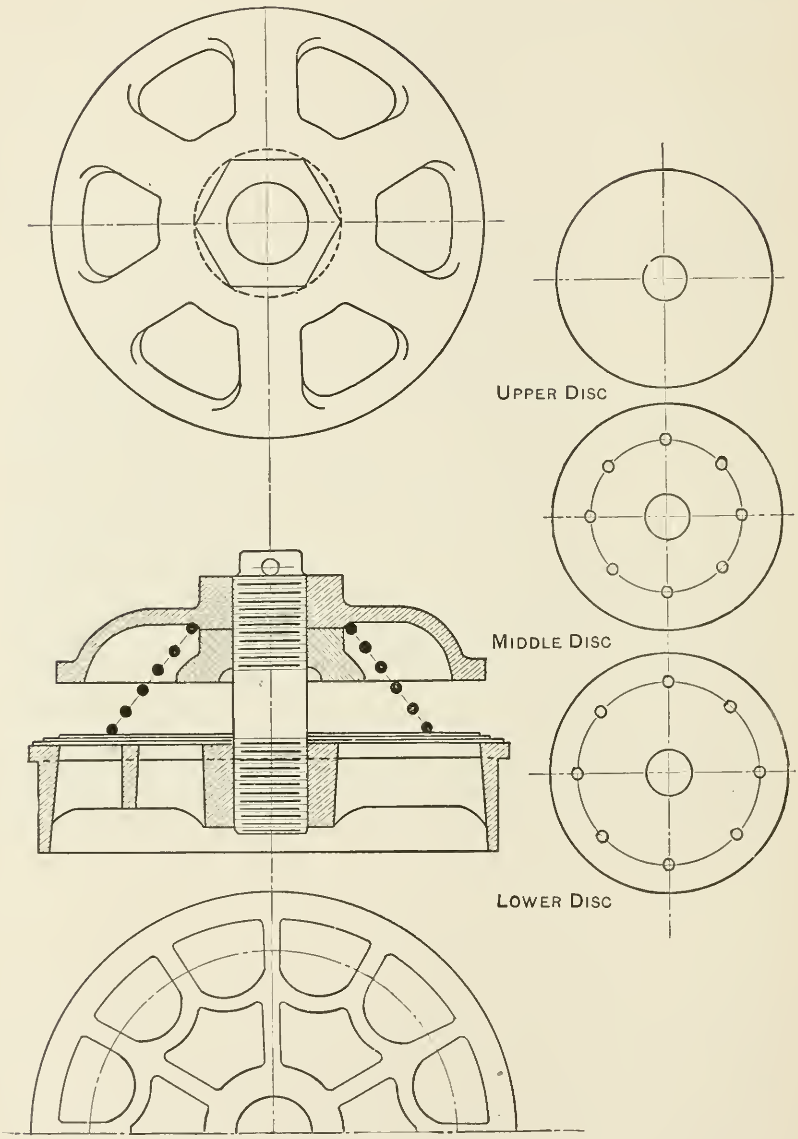


FIG. 87.

The velocity of the air and water through the air-pump suction pipe should be from 10 to 15 feet per second based upon the effective strokes of the pump. The size of the discharge pipe can be much smaller as the air is much reduced in volume when discharged from the pump.

The thickness of metal in the body and ends of the pump should be $(0.167 \times \text{diameter of pump}) + 0.25$ inch.

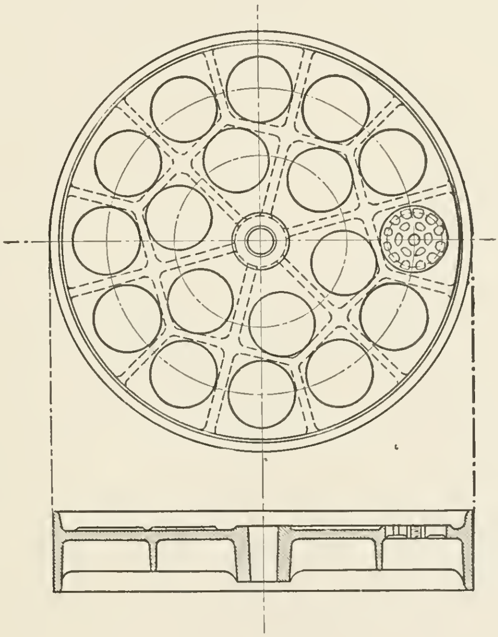


FIG. 88.

171. Types of Air Pumps. — Air pumps can be divided roughly into four classes:

1. Reciprocating pumps with three sets of valves.
2. Reciprocating pumps with one set of valves.
3. Compound pumps.
4. Rotary pumps.

Fig. 89 is an example of a pump of class (1), with a set of head valves, bucket valves, and foot valves. Fig. 90 shows a set of indicator cards taken from such a pump. It will be seen that

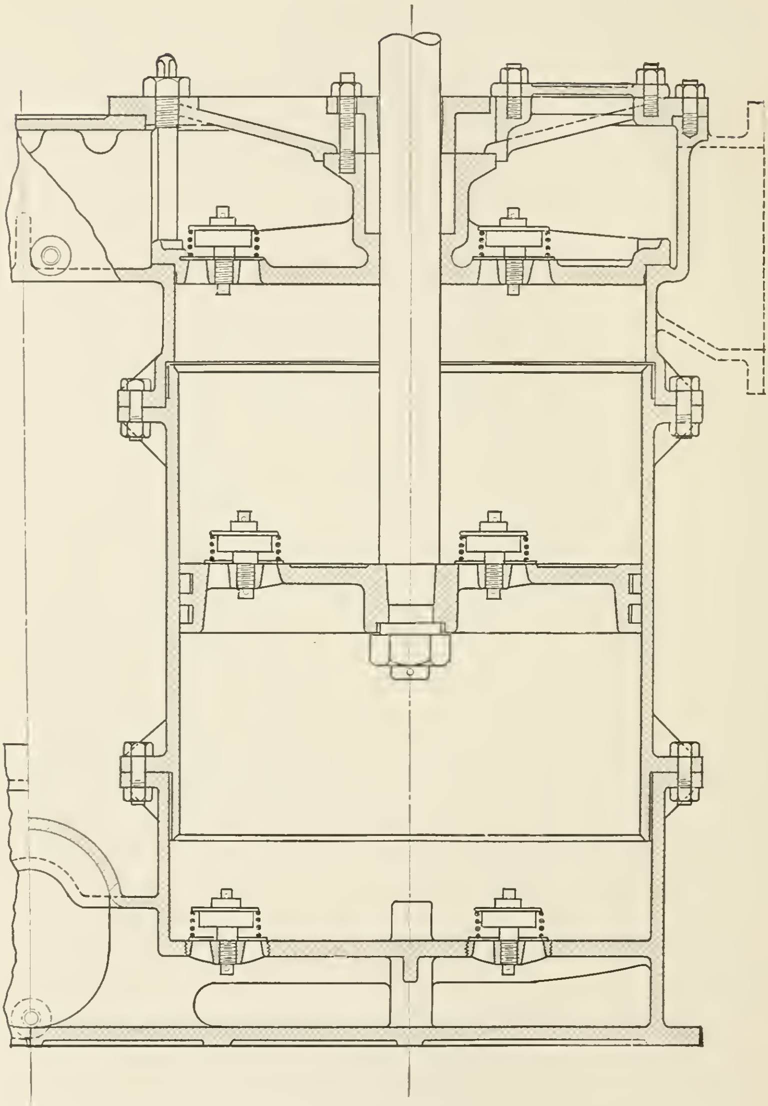


Fig. 89.

the pump works in two stages. On the under side of the bucket the pressure varies from the vacuum to a pressure about half-way between that and the discharge pressure. On the upper side of the bucket the pressure varies from something less than the upper pressure of the under side to the discharge pressure of the upper side. The discharge pressure will depend upon whether the pump is discharging into a hotwell, as with a surface condenser, or overboard, as with a jet condenser. The cards shown are for pumps in connection with jet condensers.

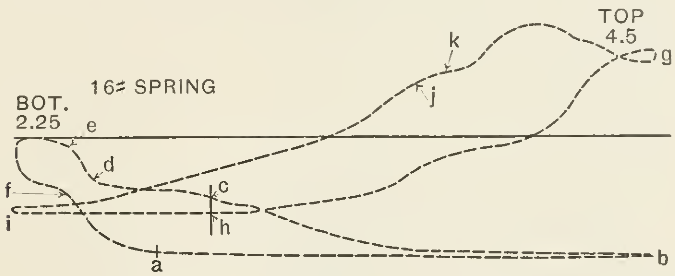


Fig. 90.

The lower part of the air pump is in communication with the condensers from about *a* to *b* (Fig. 90), at which point the bucket is at the top of its stroke. As it starts down the foot valves close and the pressure increases from *b* to *c*. At the same time the pressure on the top of the bucket drops from *g* to *h*. When there is a sufficient pressure difference the bucket valves open and air passes through the valves from *c* to *d*. At that point the bucket strikes the water and the increased resistance to its passage through valves causes the pressure to rise to *e*. As the bucket starts upward the pressure on the under side at first drops rapid y as the valve closes and then remains nearly constant up to *f*, while the air which has been forced into the water in the clearance space frees itself. The events on the top side can be easily followed and we find the same sudden increase at *k* when the water passes out, and the same slowly falling pressure due to air in the water in the clearance space. These cards show very plainly that even with water in the clearance spaces the bad effect of clearance is not entirely obviated.

The Edwards pump, see Fig. 91, is an example of an air pump of class (2). The pressure on the under side of the pump is practically constant and equal to the vacuum, while on the top the pressure varies from that of the vacuum to something above atmospheric pressure. The action of a pump of class (1) is more or less intermittent. There is not enough difference in pressure between the condenser and the pump barrel to cause the water to flow through the foot valves rapidly. It will accumulate until the pressure due to its own head and that of the air forces it into the pump, which for a few strokes will pump only water, and then only air while more water accumulates. In the Edwards pump the flow of water does not depend upon the pressure difference but the water is caught between the under side of the bucket and the bottom

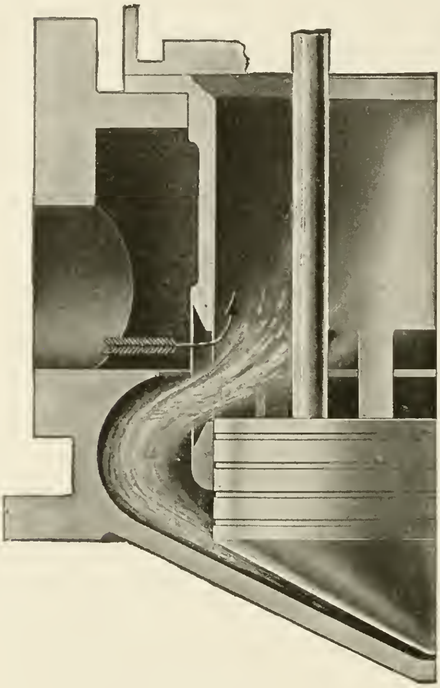


Fig. 91.

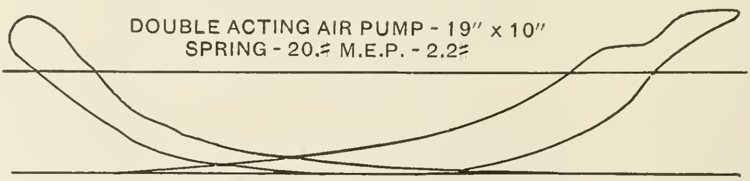


Fig. 92.

end of the pump and forced onto the top of the bucket. In Fig. 76 is shown a double-acting pump of class (2). Fig. 92 shows a set of cards taken from this pump. In the top card we see the

sudden increase in pressure when the water passes through the top valves. On the under side of the pump the water passes

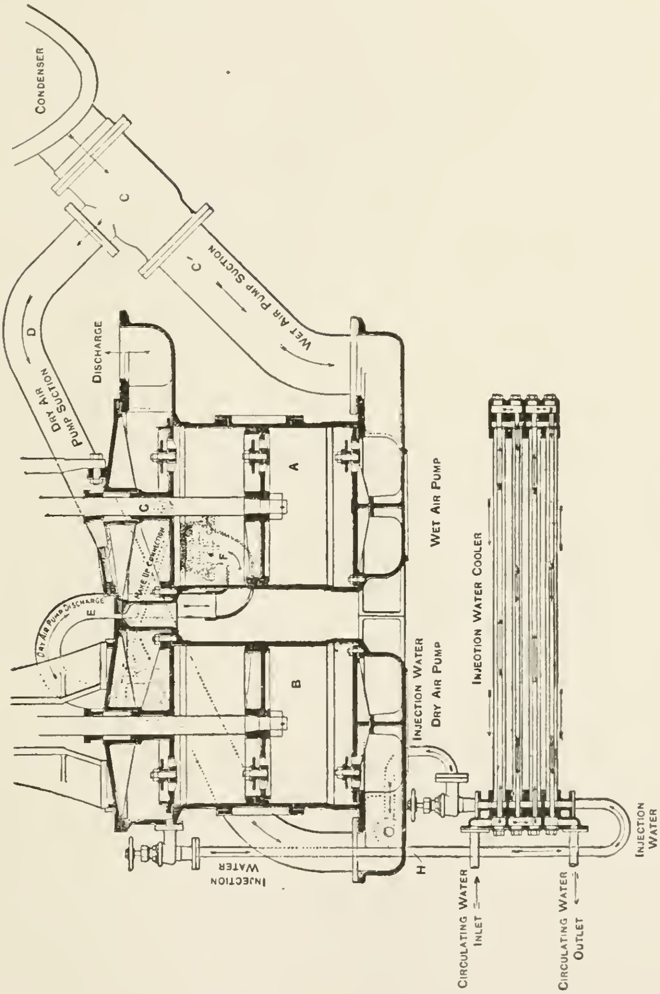


FIG. 93.

out first, then the air, causing the pressure to run up to a maximum and then drop as the air passes out.

In Fig. 90 it will be seen that the pump is open to the condenser for only 0.75 of the stroke, due to the slowness of the bucket valves in closing and more especially to the escape of air from

the water in the clearance space. These same conditions reduce the time that the bucket valves and head valves are open to about 0.3 of the stroke. These diagrams show clearly the harmful effect of clearance and high delivery pressure.

In maintaining a vacuum of 28 inches or more with pump of class 1 or 2, the clearance must be extremely small or else the operation must be performed in a greater number of stages. This increase in the number of stages can be effected by using two Edwards pumps, one delivering to the other. The arrangement shown by Fig. 93 is used by J. G. Weir & Co., where a high vacuum is desired. It can be readily seen that the water will flow to pump *A* and the air to pump *B*. Pump *A* is under the steam cylinder as this is the pump that does the most work. The load upon *B* is slight and it is worked by beams and links from the piston rod of *A*. A portion of the compression of the air takes place in the lower part of *B*, and it is still further compressed in the upper part of *B* to a pressure of about 5 pounds absolute. It is then delivered to the upper part of *A* where it is brought up to the atmospheric pressure and delivered. The water for cooling the dry pump is circulated through the cooler as shown, the circulation being maintained by the difference in pressure between the suction and discharge sides of the pump.

Fig. 94 shows a LeBlanc rotary air pump. In this type of pump the question of clearance is eliminated. The water is thrown towards the discharge orifice in thin sheets and carries the air out in this way. It is really a partial admission turbine running backwards and delivering energy to the water. Some tests upon this pump showed that it required about one I.H.P. for 2.2 pounds of water per second, and one volume of water would handle four volumes of air.

172. Condition for Maximum Load. — In estimating the horse-power of an air pump it should be remembered that the maximum horse-power is not required for the highest vacuum but for a back pressure of about 8 or 9 pounds. If we assume

$$\text{M.E.P.} = Pi\,\frac{1 + \log_e r}{R} - Pb,$$

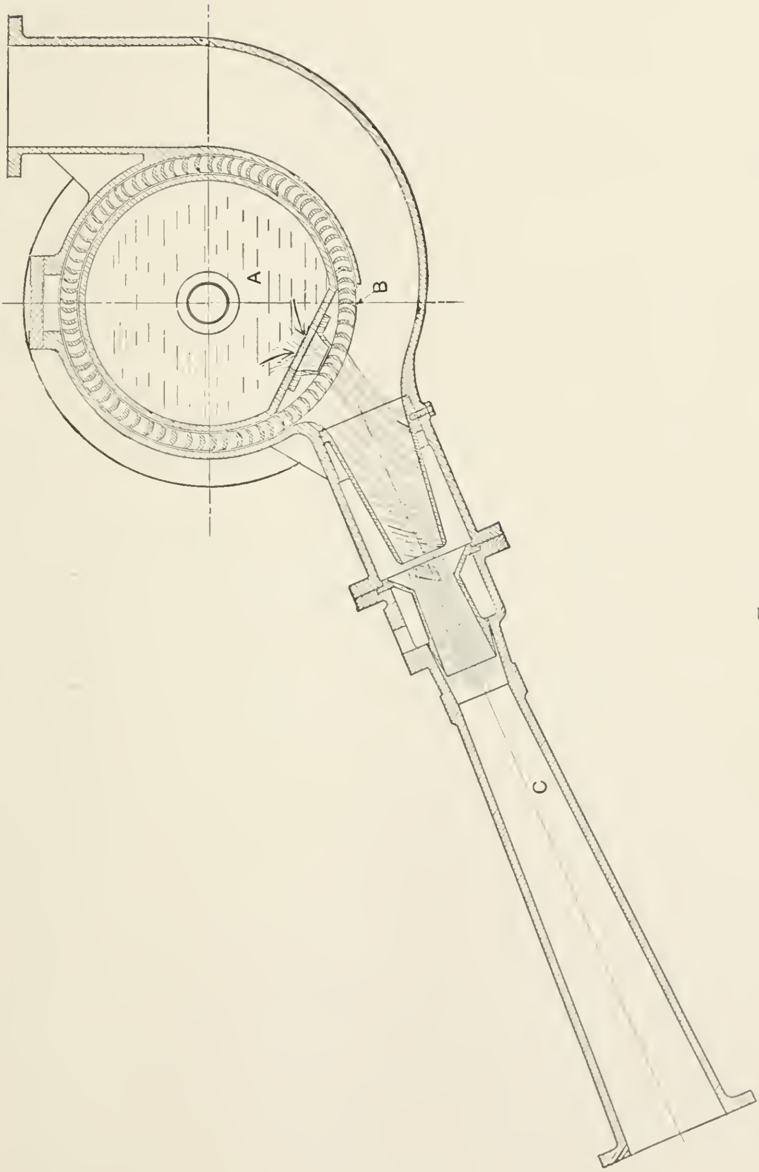


Fig 94.

and let $$R = \frac{Pi}{Pb},$$

then $$\text{M.E.P.} = Pb \log_e \frac{Pi}{Pb}.$$

If we differentiate this and put the first differential = 0 we find that the M.E.P. has its maximum value when $\frac{Pi}{Pb} = e = 2.7183$, and when this relation exists the M.E.P. = Pb. As a matter of fact the effect of clearance, etc., is such that the actual M.E.P. is about one-half the theoretical. The pump should be designed with enough power to produce this reduced vacuum, if conditions are such that the higher vacuum cannot be maintained.

A paper by Mr. William Weir in the "Transactions of the Institution of Engineers and Shipbuilders in Scotland," Oct. 22, 1912, gives the following information concerning air pumps.

Engine	Steam per I.H.P., pounds	Type of air pump	Designed vacuum, inches	Ratio of bucket displacement to volume of feed-water	Weight of pump per H.P. of main engine, lbs.
1	17	Twin	26	12.4	2.09
2	15	Dual	26	7.4	1.10
3	16	Twin	26	15.4	2.16
4	14	Dual	28.5	13.3	1.38
5	14	Dual	28.5	13.2	1.23
6	16	Monotype	26	12.7	0.765
7	14.5	Dual	28	8.8	0.466
8	....	Attached	25	60	......
9	....	Attached	25	45	......
10	....	Attached Dual	25	26.5	......

SECTION V

TURNING ENGINES AND REVERSING ENGINES

TURNING ENGINES

173. Type of Engine. — The turning engine is a small engine of one or two cylinders used for turning the main engine over when the latter is being overhauled and when the valves are being set. The engine is designed to turn the main engine over once in from five to ten minutes and since the r.p.m. of the engine are from 200 to 400, the turning engine makes from 1000 to 4000 revolutions while the main engine is turned over once. The diameter of the cylinder, or cylinders, and the length of the stroke are usually from 6 to 8 inches. The most convenient means of making this large reduction in revolutions is by use of worms and worm-wheels. In engines of 1000 I.H.P. or less, one worm and wheel can be used and in larger engines two sets are employed. Small engines of 400 I.H.P. or less are usually turned over by means of a bar.

If the worm is single threaded the worm wheel will advance one tooth for each revolution of the worm, and with the arrangement shown in Plate 1 it will be readily seen that the following relation will hold:

$$rm = nn_1. \qquad (85)$$

r = r.p.m. of turning engine.
m = number of minutes to turn over main engine once.
n = number of teeth on small worm-wheel.
n_1 = number of teeth on large worm-wheel.

The value of rm is usually from 1000 to 2000 in engines under 5000 I.H.P.

174. Frictional Load. — The frictional load which the turning engine has to overcome is hard to determine exactly. When

the main engine is not under steam the cylinder walls are dry and unlubricated and the bearings are not well oiled. The speed at which the main engine turns is so very slow that the coefficient of friction will be practically that due to starting a body from a state of rest. Experiments have shown that it takes about 2.5 pounds mean effective pressure on the L.P. cylinder to overcome engine friction when running at a piston speed of 150 to 200 feet per minute. If we assume that the coefficient of friction from a state of rest is about four times the coefficient of friction at 200 feet per minute, and that the dry condition of bearings and cylinders will increase the frictional load about 50 per cent, we shall have the frictional load to be overcome in one revolution of the main engine as follows:

$$A = \frac{\pi D^2}{4} \times 2\,S \times \text{M.E.P.}_{f} \times 4 \times 1.5. \tag{86}$$

D = diameter of L.P. cylinder of main engine in inches.

S = stroke of main engine in inches.

M.E.P.$_{f}$ = about 2.5.

175. Power of Turning Engine. — The power of the turning engine, driving through the worm and wheels, will vary according to the number of teeth on the worm-wheels, the efficiency of the engine and gearing, and the pressure of the steam. The engine is usually run from the auxiliary steam line where the pressure will be 100 pounds, gage, or less. The power finally delivered to the main engine shaft will be as follows:

$$B = \frac{\pi d^2}{4} \times 2\,s \times \text{m.e.p.} \times n \times n_1 \times e \times e_1 \times e_2. \tag{87}$$

d = diameter of cylinder of turning engine in inches.

s = stroke of turning engine in inches.

m.e.p. = mean effective pressure in cylinder = about 0.75 P.

P = steam pressure at engine.

n and n_1 are as above.

e = mechanical efficiency of engine = about 0.8.

e_1 = efficiency of small worm and wheel = about 0.4.

e_2 = efficiency of large worm and wheel = about 0.4.

Equating A and B and substituting the values for efficiency, etc., the following equation is obtained:

$$d^2s = \frac{D^2 \times \text{M.E.P.}_f \times S \times 63}{n \times n_1 \times P}. \tag{88}$$

If special attention is paid to the design of the worm and wheel, and if the worm runs in a bath of oil and has a thrust bearing, the efficiency will be higher than 0.4, but as ordinarily designed for turning engines the gears will have about that efficiency.

176. Proportions of Teeth of Worm and Wheel. — The teeth of the worm and wheel must be figured to stand the stress resulting from the transmission of power, and in order that they may have a practical thickness at the root it is usual to employ a pitch of 1.75 to 2.25 inches on the small worm-wheel, and of 2.25 to 3.5 inches on the large worm-wheel. The proportions of the teeth should be about as follows:

Length of teeth = 0.65 pitch.
Face of teeth = 0.3 pitch.
Flank of teeth = 0.35 pitch.
Thickness of teeth at pitch circle = 0.48 pitch.
Breadth of teeth at root of worm wheel = 2 to 2.25 pitch.
Least number of teeth on small worm-wheel about 25.

177. Design of Worm and Wheel. — Diameter of pitch circle of large worm-wheel = CS

$$C = 1.1 \text{ to } 1.5.$$

Let Z = inch-pounds of work done by turning engine in one stroke $= 0.75\, P \dfrac{\pi d^2}{4} s.$ (89)

The force acting upon the teeth of the small worm-wheel at the pitch circle will be

$$F = 0.75\, P \frac{\pi d^2}{4}\, 2\, s \times (0.8) \times \frac{1}{p} \times (0.6) = \frac{0.96\, Z}{p}. \tag{90}$$

p = pitch of teeth on small worm-wheel in inches.

It will be noticed in the above equation that the force acting upon the teeth is taken as 0.6 of the power delivered by the engine, although the efficiency of the gear is taken as 0.4. It is assumed that there are three sources of frictional loss, the end thrust of the worm, the friction in the bearing of the worm-wheel, and the friction between the teeth. It is assumed that each loss amounts to 0.2 of the total power delivered to the worm. Upon this assumption the power delivered to the teeth will be 0.6 of the power delivered to the worm.

The force acting upon the teeth of the large worm-wheel at the pitch circle will be

$$F_1 = \frac{2}{3} F \times \frac{pn}{p_1} \times (0.6) = \frac{0.384\, Zn}{p_1}. \tag{91}$$

p_1 = pitch of teeth on large worm-wheel.

Assume that the force acting at the pitch circle is carried by two teeth, that the teeth have the proportions given above, and that if the thickness of the teeth at the pitch circle is 0.48 p the thickness at the root will be at least 0.5 p. The stress at the root of the teeth of the small worm-wheel will be

$$f = \frac{Fl\,6}{2\,bt^2} = \frac{4.2\,F}{C_1 p^2}. \tag{92}$$

$b = C_1 p$ = breadth of teeth at root.

C_1 varies from 2 to 2.5.

l = length of teeth below pitch circle = 0.35 p.

$t = 0.5\, p$.

The stress at the root of the teeth of the large worm-wheel will be

$$f_1 = \frac{F_1 l_1\, 6}{2\, b_1 t_1^2} = \frac{2.91\, F_1}{C_2 p_1^2}. \tag{93}$$

l_1 and b_1 are the same as l and b.

$t_1 = 0.6\, p_1$ when thickness of tooth at pitch circle = 0.48 p_1.

C_2 varies from 2 to 2.50.

The stresses allowed are 3500 to 4000 pounds for cast iron, and 5000 pounds for cast steel. It is advisable to make the worm of steel and the worm-wheel of cast iron. If the worm-

wheel is made of cast steel the worm should be made of bronze. If the teeth of the worm and wheel are of equal thickness at the pitch circle the teeth of the worm do not need to be figured as they will be thicker at the root than the teeth of the wheel.

The efficiency of worm gearing improves as the pitch angle of the worm becomes larger, and in order that the angle may be as large as possible for a given pitch the diameter of the worm should be as small as possible. The worms are usually made separate and are keyed to the shaft, in which case the diameter of the pitch cylinder will be about $3 \times$ pitch. This diameter allows for a thickness of metal below the root circle of about $0.5 \times$ pitch. If the worm is cut from the shaft forging the diameter of the pitch cylinder can be about $2.5 \times$ pitch of the teeth. The length of the worm should be from 3 to $4 \times$ pitch.

The indicated horse-power of the turning engine will be

$$\text{i.h.p.} = \frac{Z \times r}{198{,}000}. \tag{94}$$

The mean turning moment on the crank shaft will be

$$M = \frac{Z}{\pi}.$$

The maximum turning moment will be about $2\,M$.

The bending moment on the crank shaft will be about 0.5 the maximum twisting moment. The diameter of the crank shaft can be determined by means of the formula:

$$\text{diameter of shaft} = C_s \times 1.72 \sqrt[3]{\frac{M}{f}}. \tag{95}$$

M = maximum twisting moment.

f = allowable stress.

C_s has values depending upon the ratio of bending moment to twisting moment.

$$\frac{\text{bending moment}}{\text{twisting moment}} = 0.25–0.50–0.75–1.0–1.25–1.5–1.75–2.$$

$$C_s = 1.09–1.17–1.26–1.34–1.42–1.49–1.56–1.62.$$

The mean twisting moment upon the shaft of the large worm will be

$$M_1 = \frac{Fnp}{2\pi} = 1.53\,Zn. \tag{96}$$

The maximum twisting moment will be about $1.25\,M_1$ as the flywheel on the crank shaft will cause the turning moment of the small worm to be more uniform. The small worm-wheel usually overhangs the bearing, so there will be a bending moment upon the shaft of the large worm. The ratio of bending moment to twisting moment will be $\frac{\text{half breadth of small wheel}}{\text{radius of small wheel.}}$.

Formula (95) given above will serve for finding the diameter of this shaft also.

The mean twisting moment upon the main engine shaft will be

$$M_2 = \frac{2}{3}F_1\frac{n_1p}{2\pi} = 0.041\,Znn_1. \tag{97}$$

The relation between the number of teeth and the other proportion of the large worm-wheel will be as follows:

$$f_1 = \frac{2.91\,F_1}{C_2p_1^2}.$$

$$F_1 = \frac{0.384\,Zn}{p_1}, \text{ and } p_1 = \frac{C\pi S}{n_1}.$$

$$\therefore \quad f_1 = \frac{Znn_1^3}{27.8\,C_2C^3S^3}, \tag{98}$$

or

$$nn_1^3 = \frac{27.8\,f_1C_2C^3S^3}{Z}.$$

$$nn_1 = rm.$$

$$\therefore \quad n_1^2 = \frac{27.8\,f_1C_2C^3S^3}{Zrm}. \tag{99}$$

$$n = \frac{rm}{n_1}. \tag{100}$$

n should not be less than 25.

The large worm-wheel is usually made in halves so the number

of teeth on the wheel must be even. The quantities should be determined in the following order:

Assume r, m, M.E.P.$_f$, and P.

Find d, s, and Z. Use Formulæ (88) and (89).

Assume f_1, C_2, and C (these values may be changed later).

Find n_1 and n. Use Formulæ (99) and (100).

Find p_1 (from n_1 and C) and F_1 [from Formula (91)].

Find values of C, C_2, and f_1 (assumed before) which agree with results already found.

Find diameter of pitch circle of large worm-wheel from p_1 and n_1.

Solve Formula (93) for f_1 with the finally determined values of C and C_2.

Assume p, and value of C_1 for small worm-wheel.

Find F [from Formula (90)] and f [f om Formula (92)].

Find diameter of small worm wheel from p and n.

Find i.h.p., diameter of shafts, diameters and lengths of worms.

It will be noticed that certain coefficients and quantities have to be first assumed and then determined exactly. The assumptions do not always give practical results and slight changes have to be made to get whole or even numbers of teeth, and to get the breadths to quarters of an inch.

REVERSING ENGINES

178. Types of Reversing Engines. — Reversal of motion in marine engines fitted with Stephenson's valve gears is obtained by throwing the links over, in order that the valve stems may be actuated by eccentrics properly set to give the reverse motion. In engines of 500 I.H.P. or less, the size of the parts is such that this reversal can be accomplished by hand, but in large engines the friction of the valves and stuffing boxes is such that it requires a steam engine to throw the links over quickly. There are two types of reversing engines in general use for this purpose, the direct-acting (see Plates 1, 2, and 3) and the all-round (see Fig. 95). In the first type the engine is directly connected to the reverse shaft and makes only one stroke in reversing. This

necessitates a steam cylinder whose diameter is from 0.19 to 0.2 the diameter of the L.P. cylinder. In the all-round type the power is obtained from one or two small cylinders from 6 to 8 inches in diameter and making from 200 to 400 r.p.m.

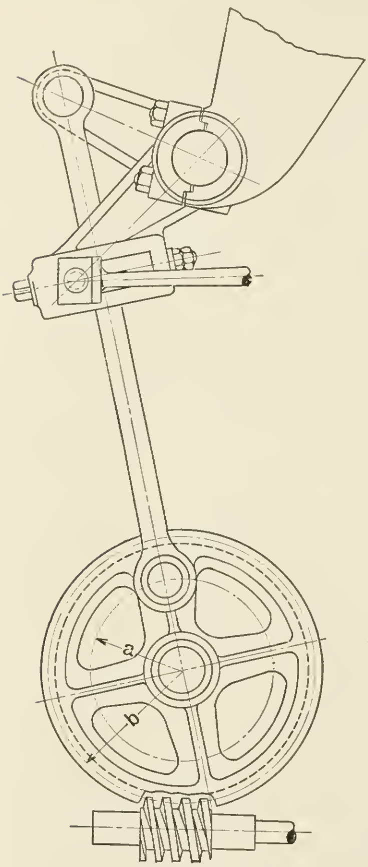


Fig. 95.

One advantage of the all-round type is that there is less danger of damage from careless handling, and another advantage is that the same engine can be used as a turning engine by a proper arrangement of gears and couplings.

179. Direct-acting Engine.— The cylinder of the direct-acting gear is supplied with steam at boiler pressure and the load upon the piston rod and connecting rod can be taken as

$$w = BP \times 0.038 \text{ L.P. area.} \tag{101}$$

BP = boiler pressure, gage.

The diameter of these rods at the middle can be found by means of Formula (12), or by Fig. 22. The diameter of the connecting rod at the ends can be 0.85 of the diameter at the middle. The length of the piston rod will be about 1.5 the stroke of the reversing engine, and the stroke of the reversing engine will be about 0.75 of the distance between the eccentric rod pins on the links. The length of the connecting rod will be from 2 to 2.5 the stroke of the reversing engine, depending upon the relative location of engine and reverse shaft.

The pin connecting the piston rod and connecting rod can be

designed to carry a bearing pressure of 3000 pounds to the square inch, since the swing of the connecting rod is very small and the engine is not in use a sufficient length of time to heat the pin. The crosshead guide is a rod of circular section and the slipper encloses it. The bearing pressure upon the slipper should be from 60 to 80 pounds per square inch. The maximum load coming upon the slipper will be

$$g = \frac{r - \sqrt{r^2 - (0.5\,s)^2}}{2 \cdot l} BP \times \frac{\pi d^2}{4}. \tag{102}$$

r = length of reversing engine lever.
s = stroke of reversing engine in inches.
l = length of reversing-engine connecting rod in inches.
BP = boiler pressure, gage.
d = diameter of reversing engine in inches.

The length of the crosshead guide will be about equal to the length of the piston rod and the diameter of the rod can be determined by considering the guide as a beam of circular section supported at the ends and loaded at the middle with the load g.

180. All-round Gear. — When the all-round gear is used the maximum pull in the drag rod (see Fig. 95) can be taken as w. The pull that must be resisted at the pitch circle of the worm wheel will be

$$\begin{aligned} A &= w\frac{a}{b} \\ &= BP \times 0.038\frac{\pi D^2}{4}\,\frac{a}{b}. \end{aligned} \tag{103}$$

a = radius of pin (Fig. 95).
b = radius of pitch circle of teeth.
D = diameter of L.P. cylinder of main engine.

If the reversing engine has a mechanical efficiency of 0.8, and if the efficiency of the worm gearing is 0.4, we shall have the power delivered to the worm wheel at the pitch circle as follows:

$$B = F \times BP \times \frac{\pi d^2}{4} \times 2\,s \times 0.8 \times 0.4 \times \frac{1}{p}. \tag{104}$$

F = ratio between the m.e.p. and the boiler pressure for the reversing engine.

d = diameter of reversing engine cylinder in inches.

s = stroke of reversing engine in inches.

p = pitch of teeth on worm-wheel in inches.

Equate A and B and we derive the following formula:

$$d^2s = \frac{0.6\,pD^2a}{Fb} \qquad (105)$$

The diameter and stroke are usually between 6 and 8 inches. The other parts of the gear, such as crank shaft, worm, and worm-wheel, can be designed in accordance with rules given under Turning Engines.

181. Cushioning Devices. — With the direct-acting gear there is danger that the cylinder heads may be carried away if the piston gets too great momentum. There are two ways of preventing this: (1) by means of an oil cylinder; (2) by means of exhaust ports placed far enough from the ends so that the piston over-travels them and a certain amount of steam is caught and compressed. In the case of the oil cylinder the flow of oil from one side of the piston to the other is controlled by a valve on the same valve spindle as the steam valve. The same motions that open and close the steam valve open and close the oil valve, and when the oil valve is closed the gear is locked in position. By throttling the oil in the passage from one side of the cylinder to the other the rate of movement of the piston can be controlled.

The valve which operates the steam cylinder may be a flat slide valve or a piston slide valve. If the latter is used the steam may be admitted either at the ends or at the middle. These valves overlap the edges of the ports by not more than $\frac{1}{16}$ inch on the steam side. Some valves are made "line and line," or with no lap at all; others are made with a clearance on the steam side so that in the "ahead" position the steam pressure will prevent the links from working over towards mid-position. In some cases the piston is cushioned by means of an exhaust lap on the valve. The lap of the oil valve should be only enough

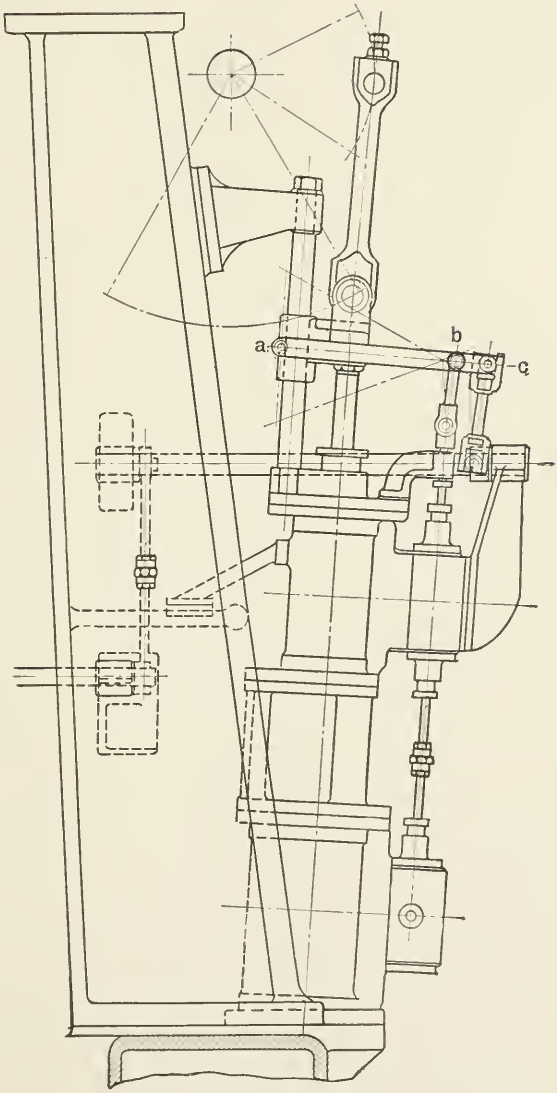


FIG. 96.

for oil tightness, as the gear will be locked as soon as this valve is closed.

182. Floating-lever Gear. — In the type of gear shown in Plates 1, 2, and 3, the valve is operated by means of a floating lever (see *ac*, Fig. 96). This lever has three points of attachment, one at *b* for the valve steam, one at *c* for the reverse lever linkage from the working platform, and one at *a* for the reverse gear. The valve gear is so set up that when the valve is moved in one direction by the reverse lever linkage the resulting movement of the reverse gear brings the valve back to its original position. The valve does not open the full width of the port, but stops are usually arranged so that the valve cannot open more than $\frac{1}{4}$ inch or so. In starting to use the reverse gear, *a* is a fixed point and a movement of *c* opens the valve. If it is desired to keep the valve open, *b* must become a fixed point and *c* must be moved by the hand lever to counteract the movement of point *a* by the reverse gear. If the reverse gear is to be stopped the reverse lever is held stationary which makes *c* a fixed point and a very slight movement of *a* closes the valve and the gear is stopped.

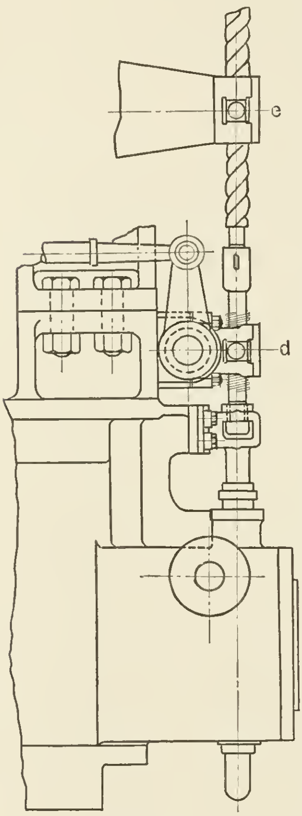


Fig. 97.

183. Brown Gear. — In the Brown gear shown in Fig. 97 the floating lever is replaced by a valve stem with a fine thread on one end and a coarse thread on the other. When the gear is to be started the valve is opened by raising or lowering the nut *d*. The nut *e* is attached to the crosshead and the movement of the nut along the thread causes the spindle to turn. If the nut *d*

were a fixed point the valve would be closed by the movement of the spindle through the nut *e*. If the nut *d* is moved up or down so as to counteract the movement of the spindle through the nut, the valve will be held open. When the gear is to be stopped the nut *d* is held fixed and a slight movement of the crosshead turns the spindle enough to close the valve.

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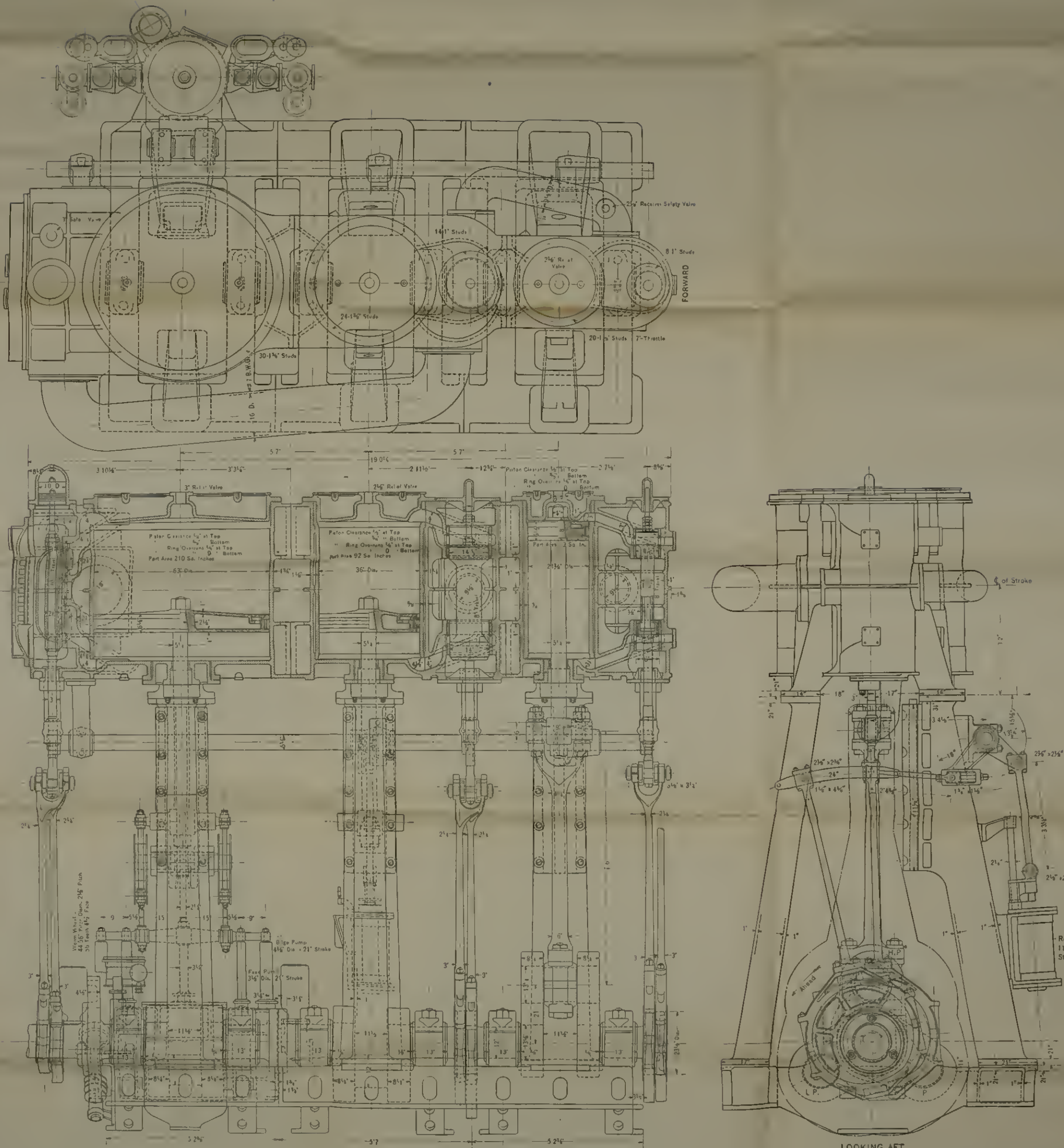
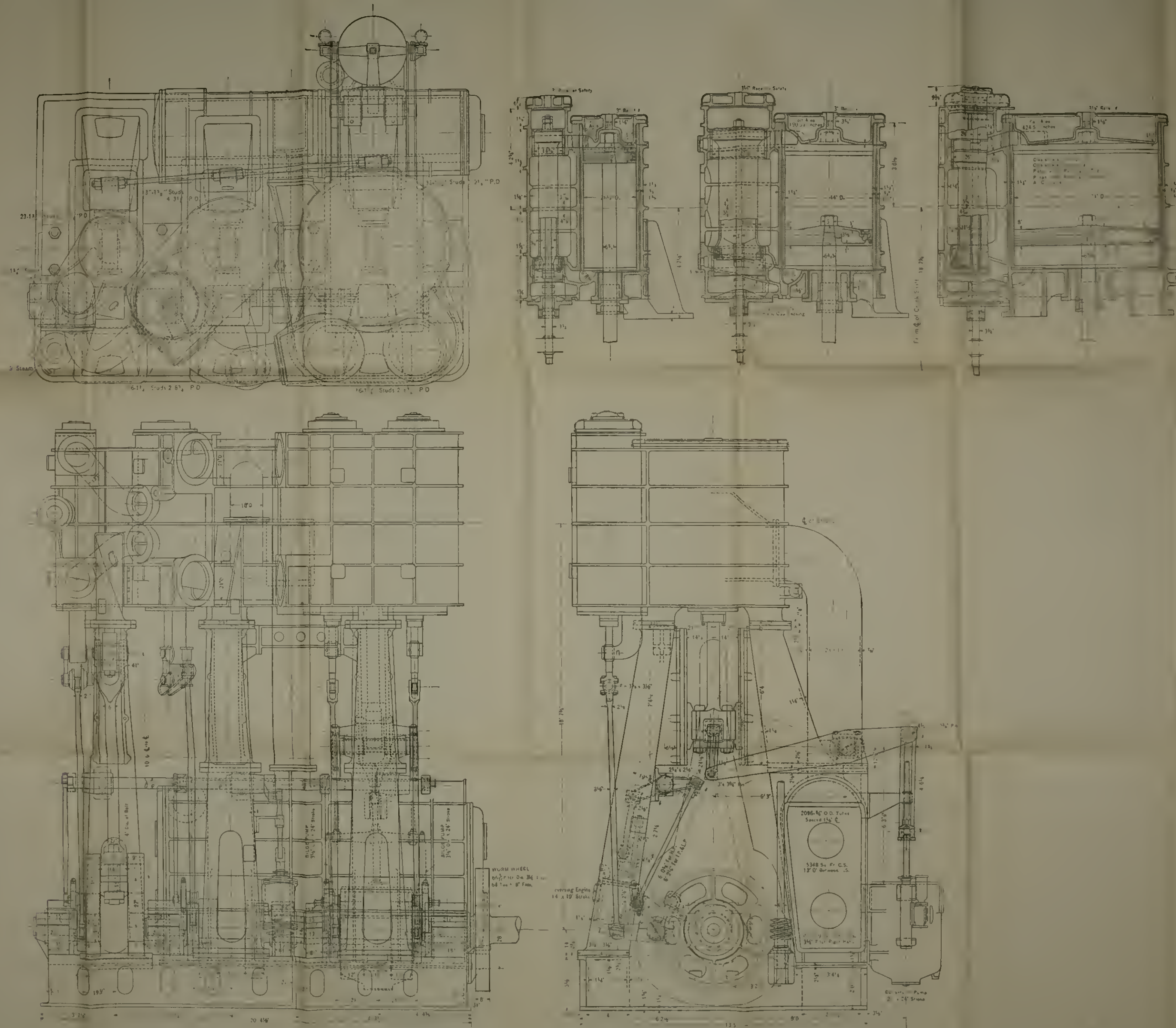


PLATE II

Designed and Built by the Newport News Shipbuilding and Dry Dock Co., Newport News, Va.



LOOKING FORWARD

PLATE III

Designed and Built by the Newport News Shipbuilding
and Dry Dock Co., Newport News, Va.

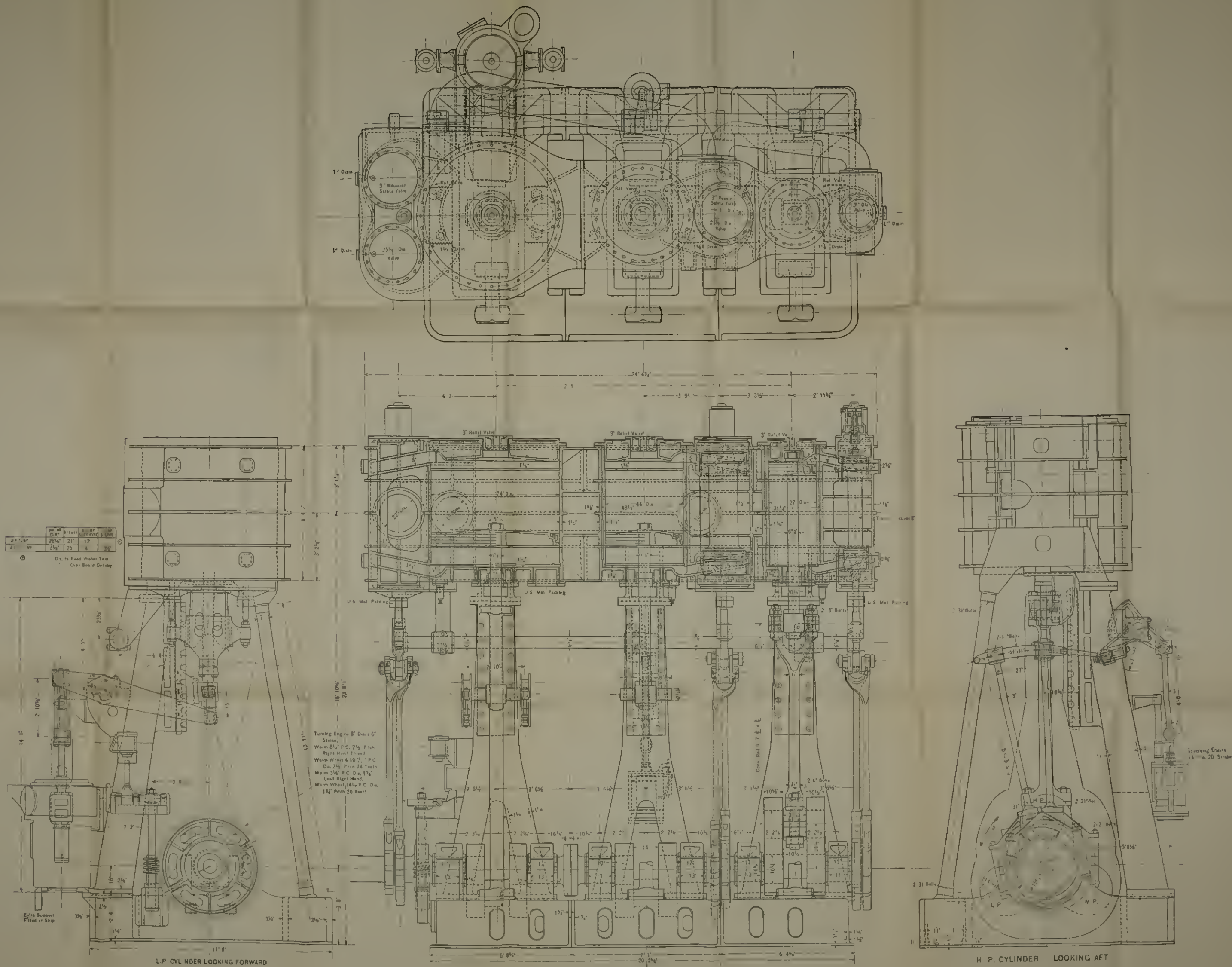


PLATE I

Designed and Built by the Newport News Shipbuilding and Dry Dock Co., Newport News, Va.

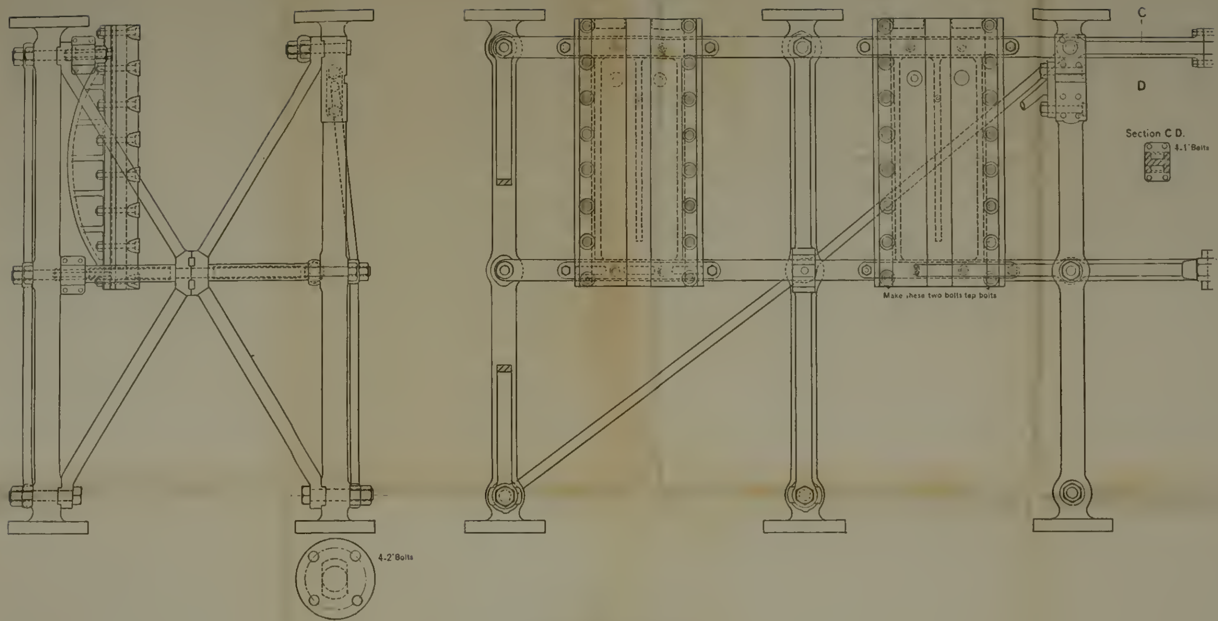

C
D
Section C D.
4-1" Bolts
Make these two bolts tap bolts
4-2" Bolts

PLATE IV